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DESIGN · PRODUCTION · MATERIALS

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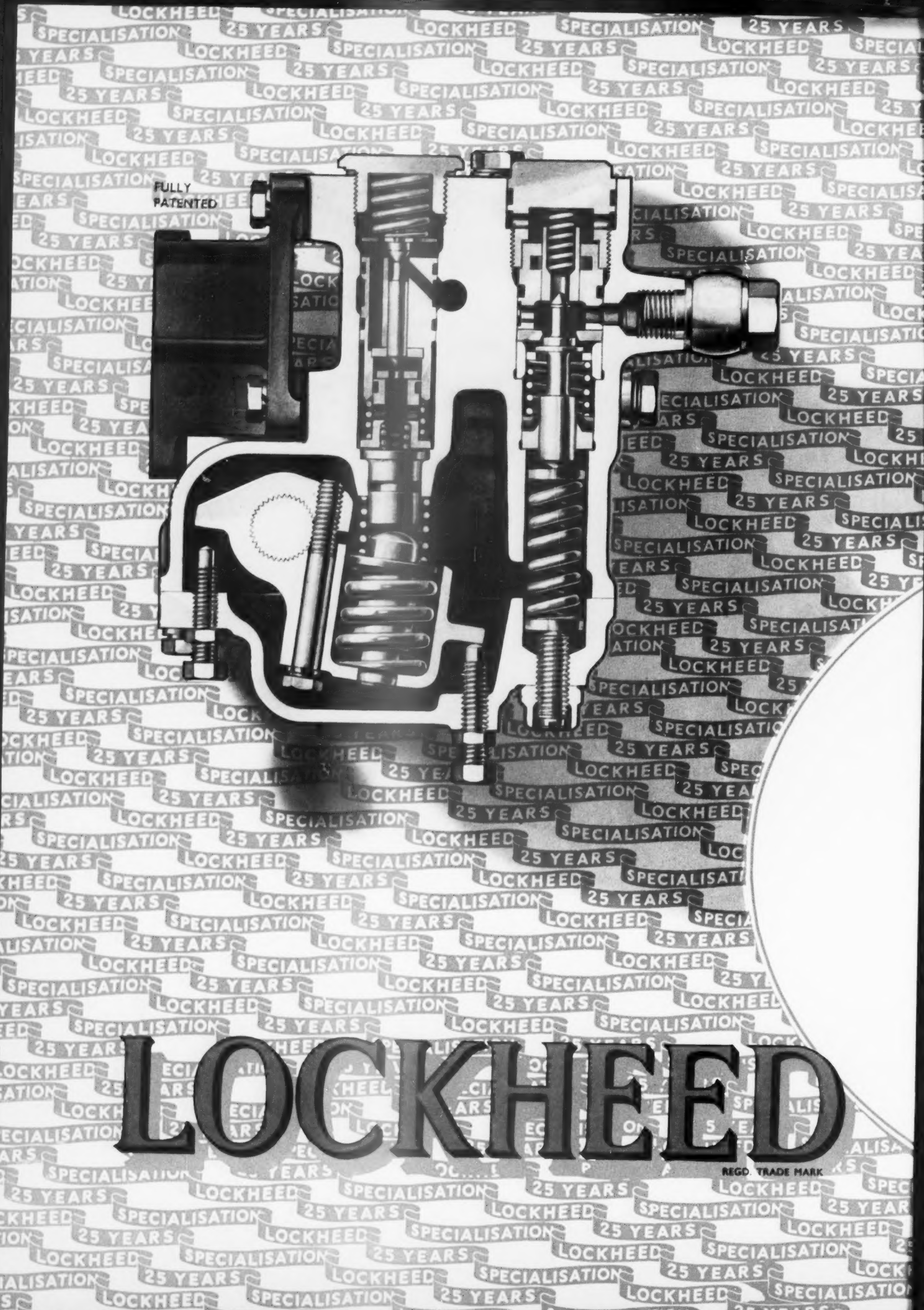
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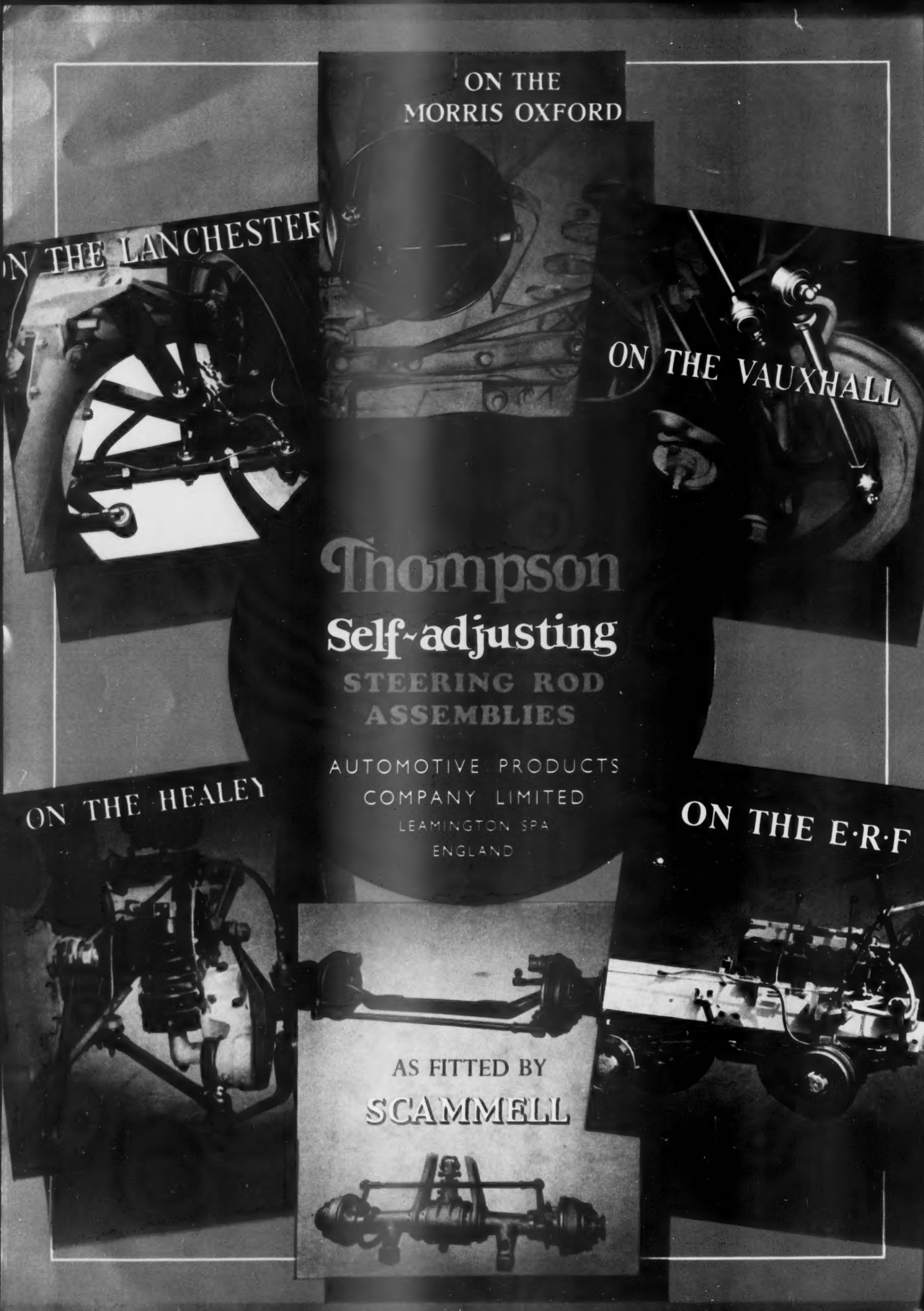

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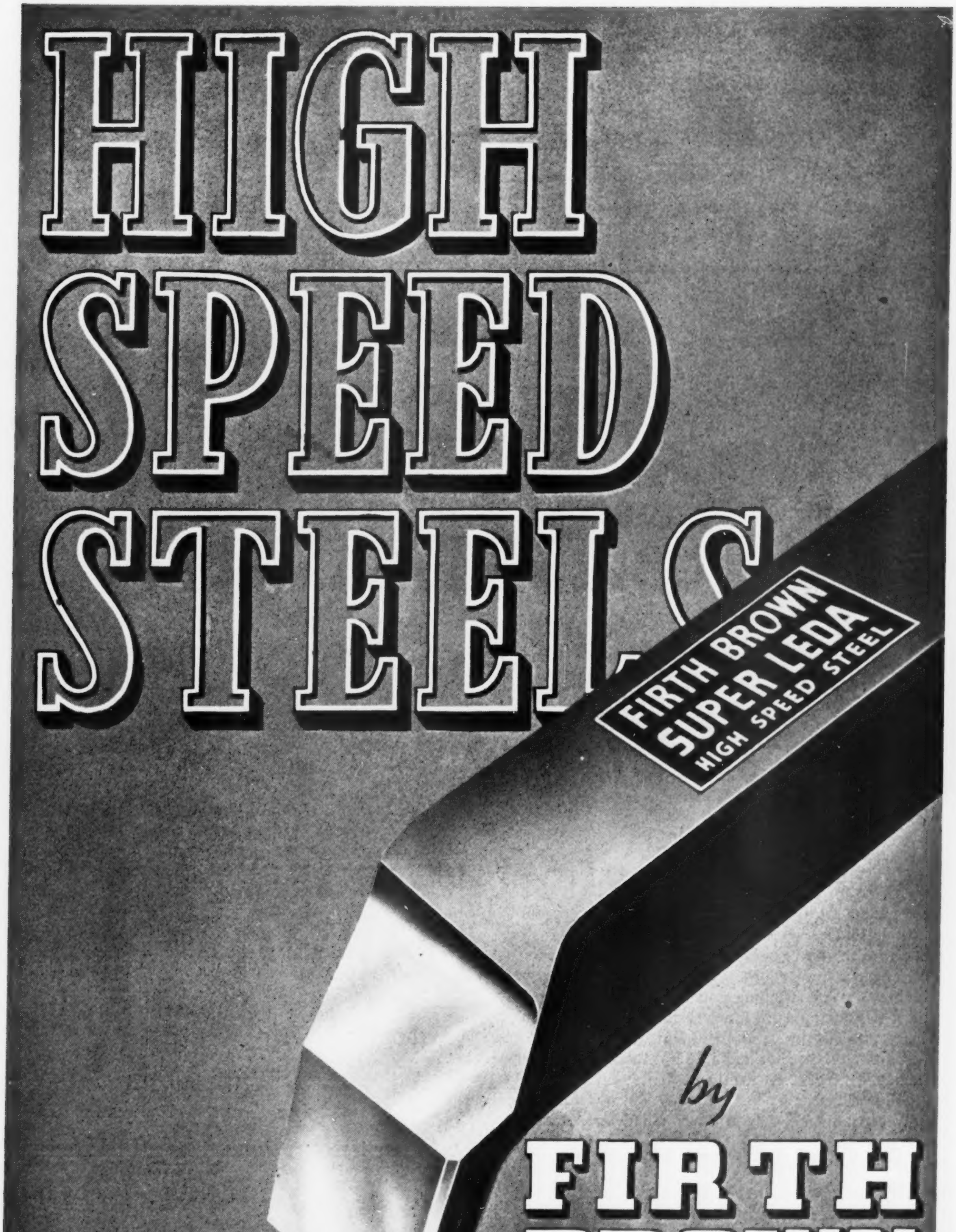
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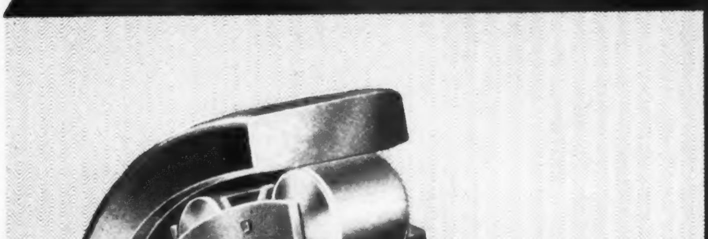

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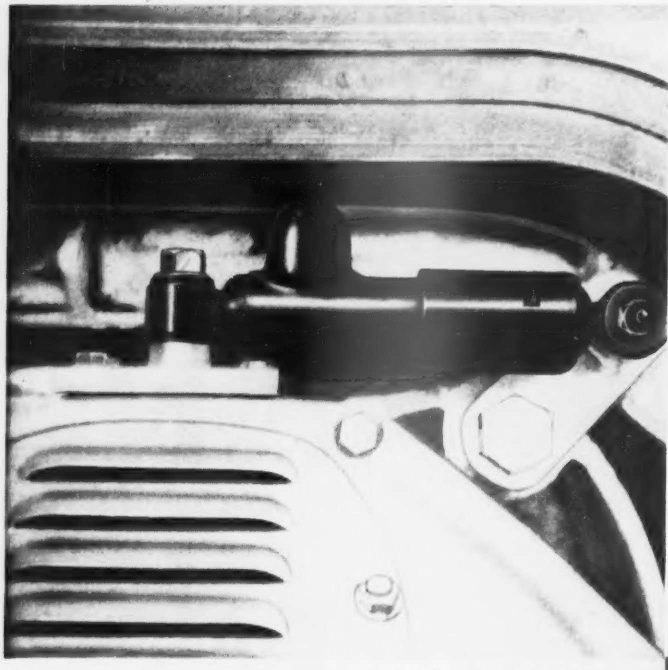

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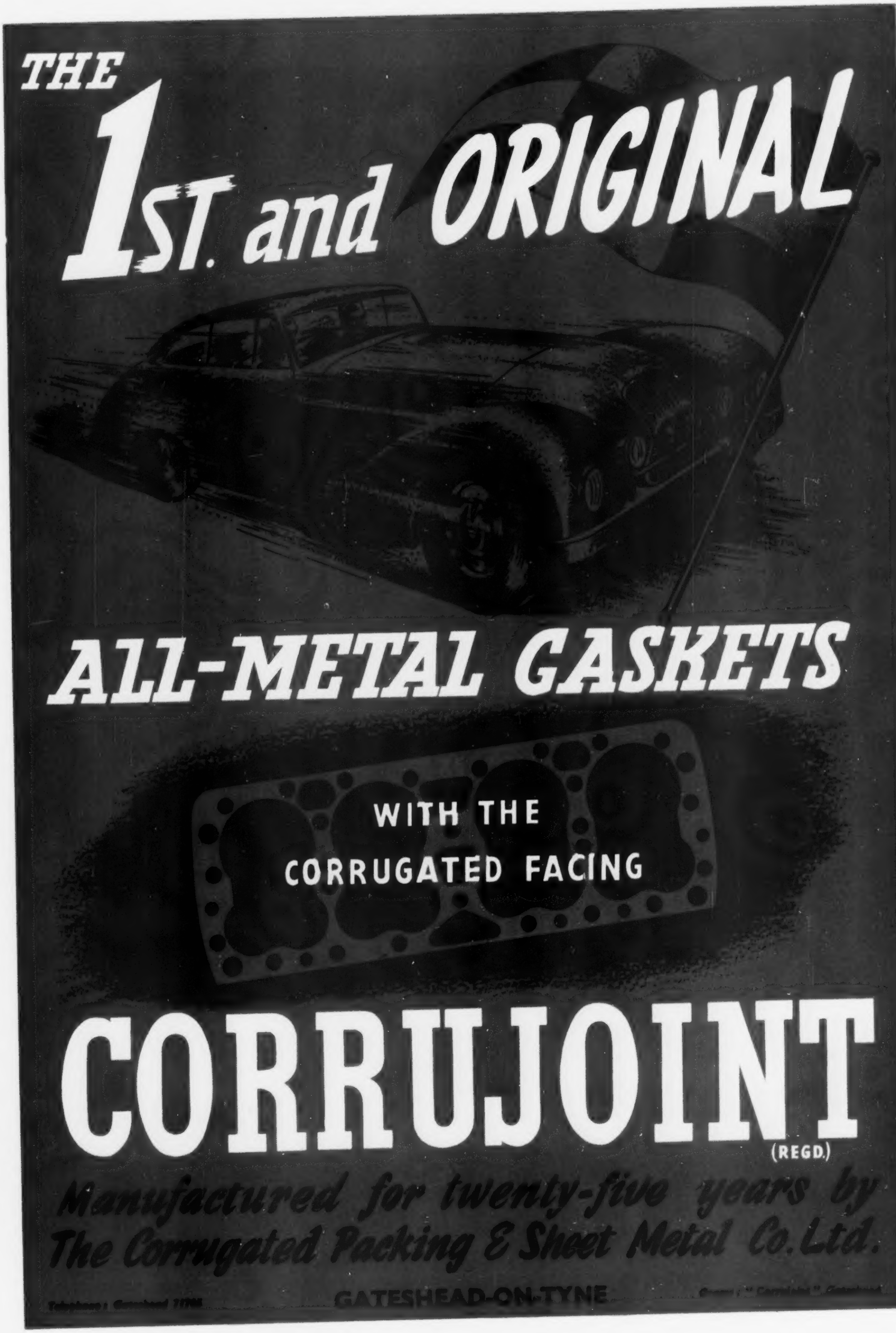
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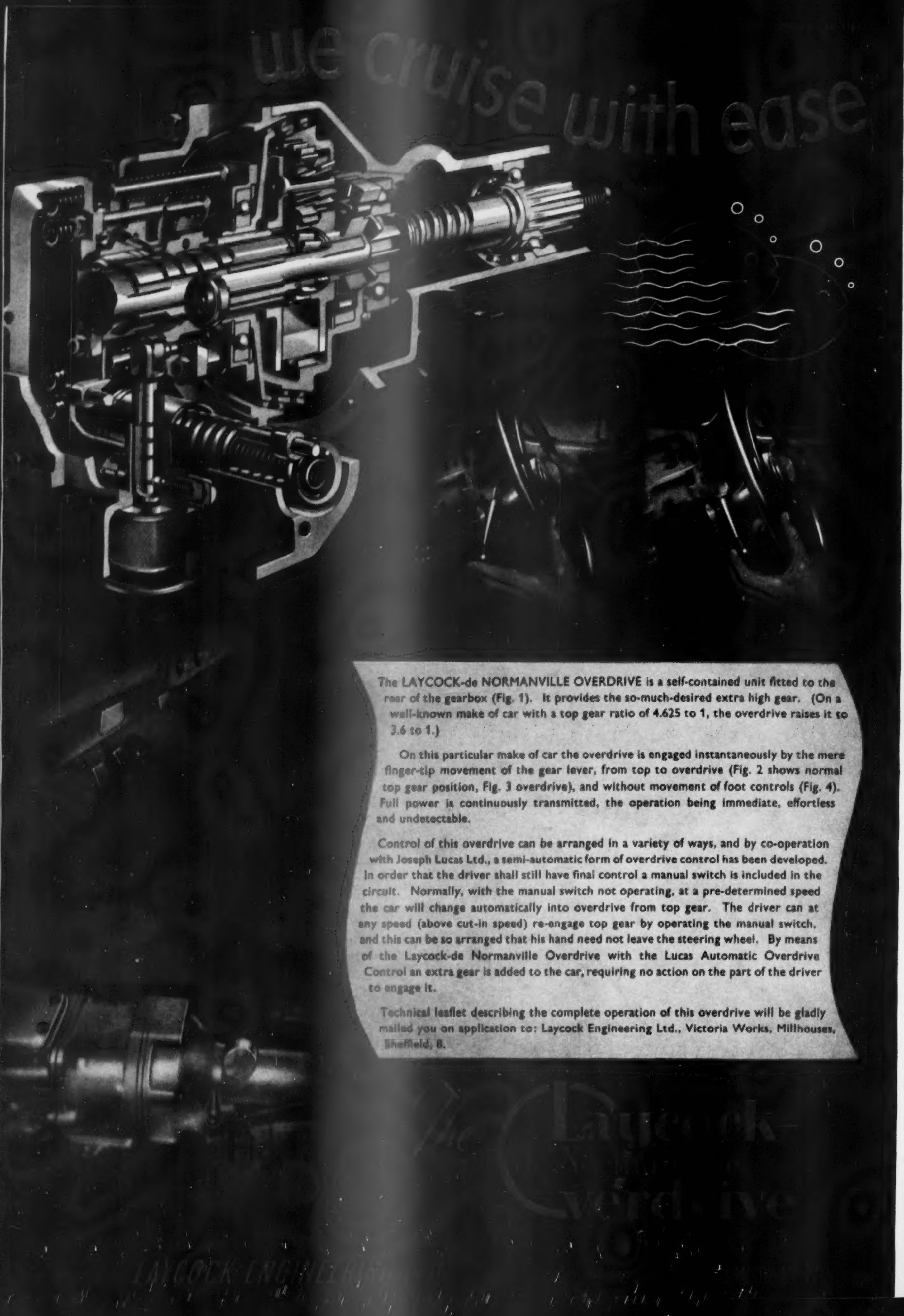
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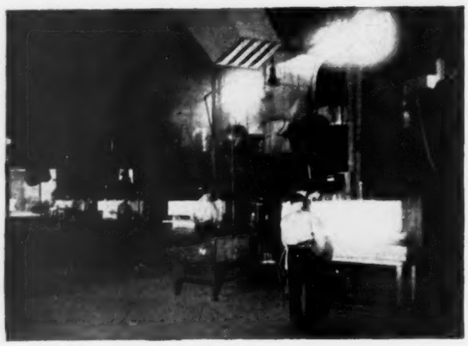

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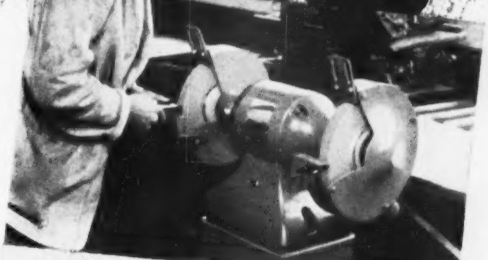

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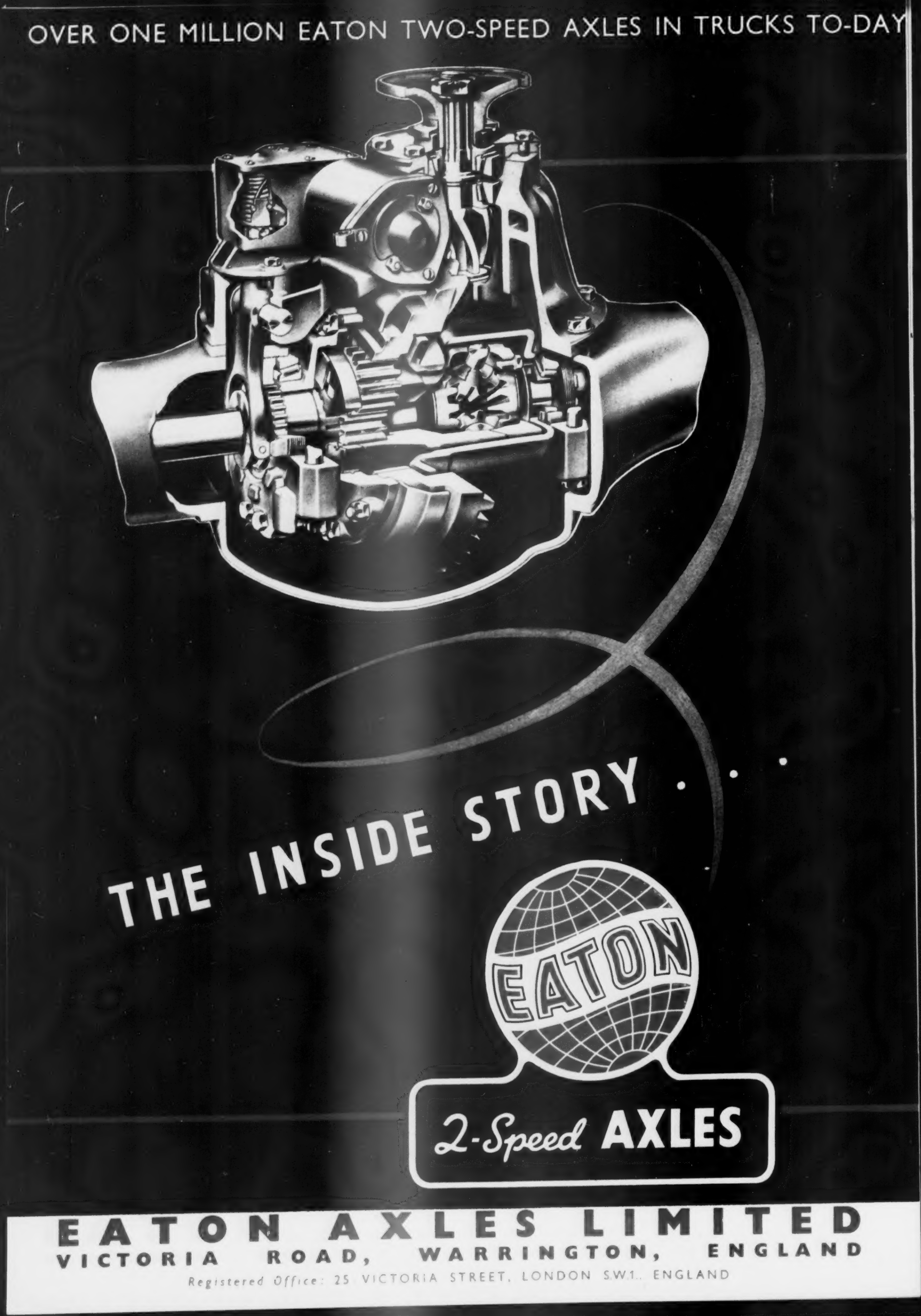
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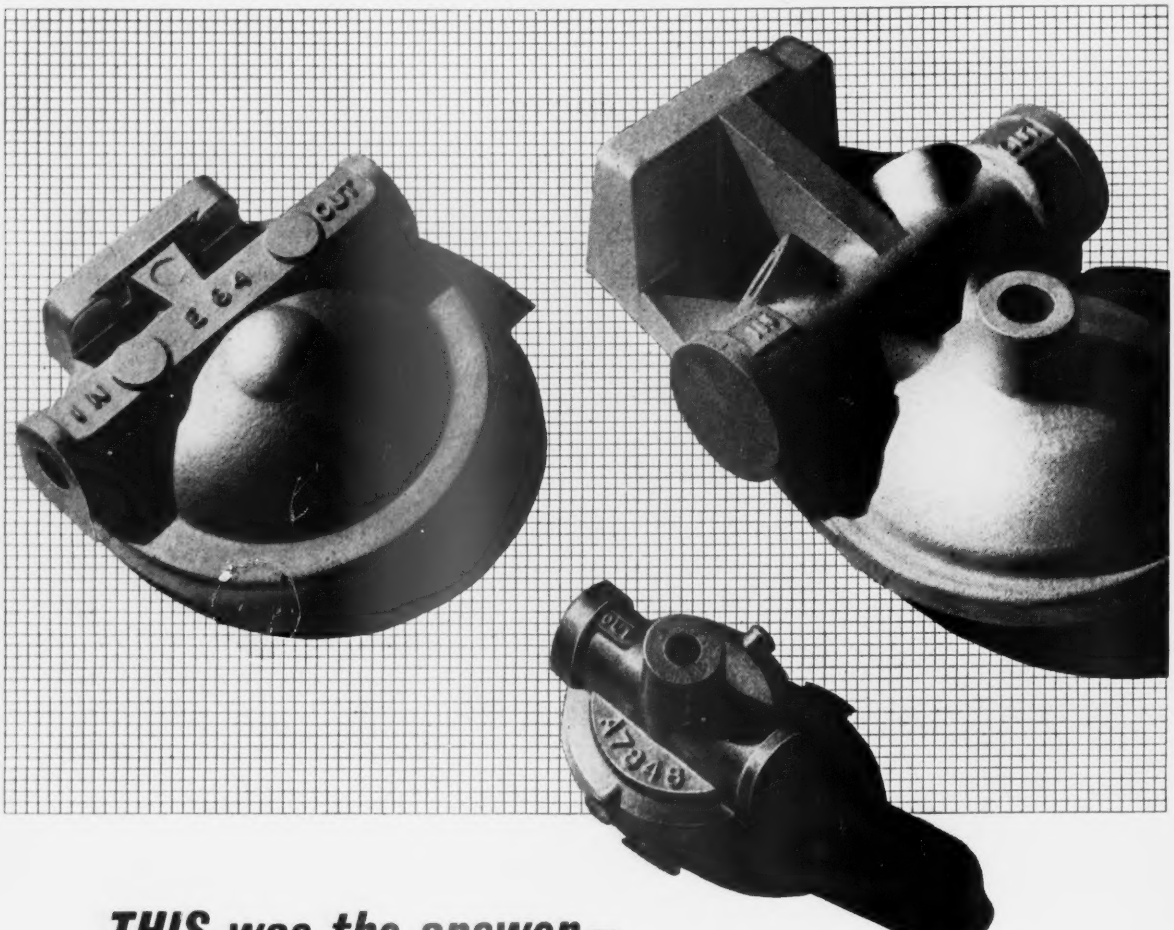

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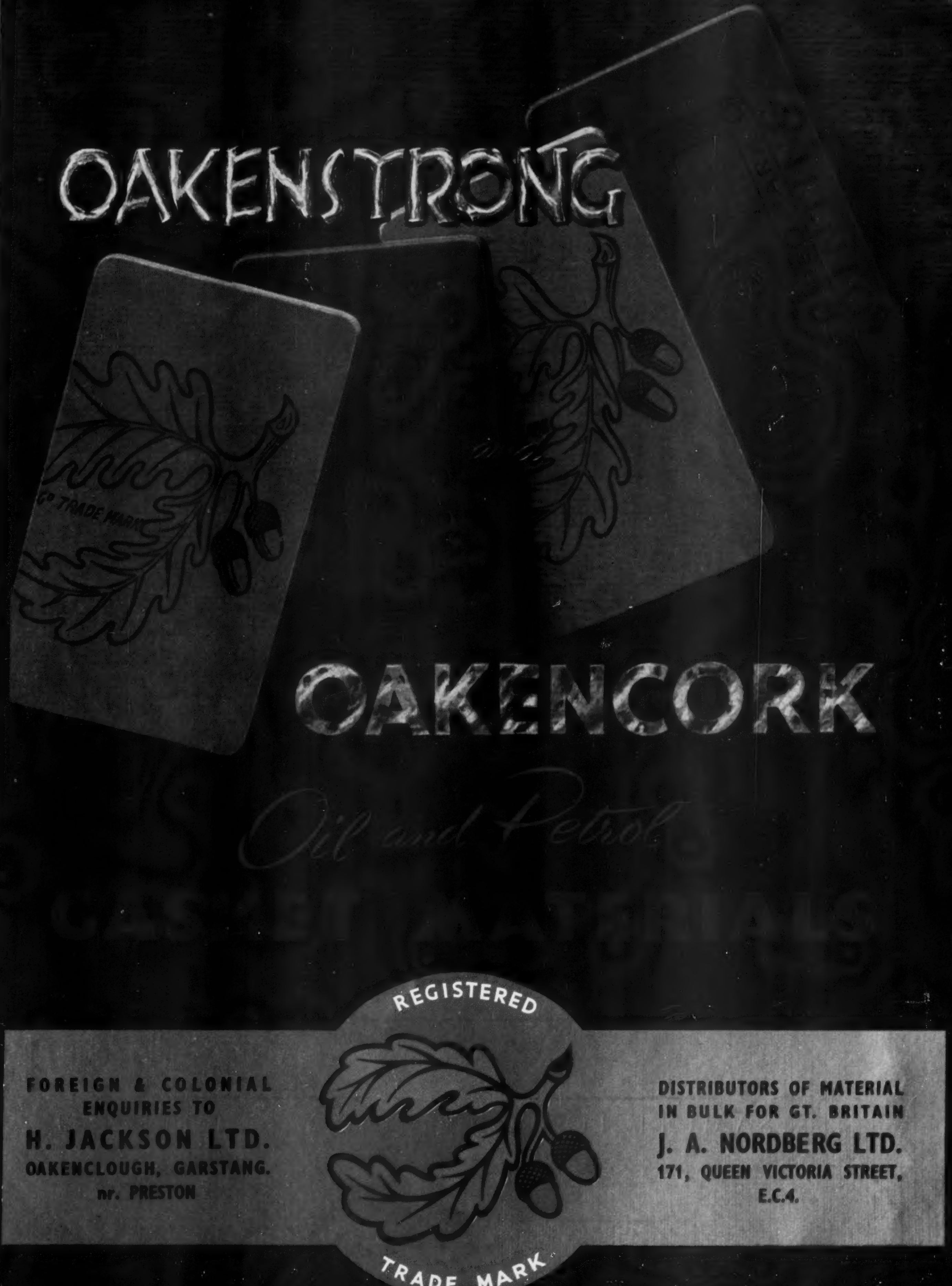
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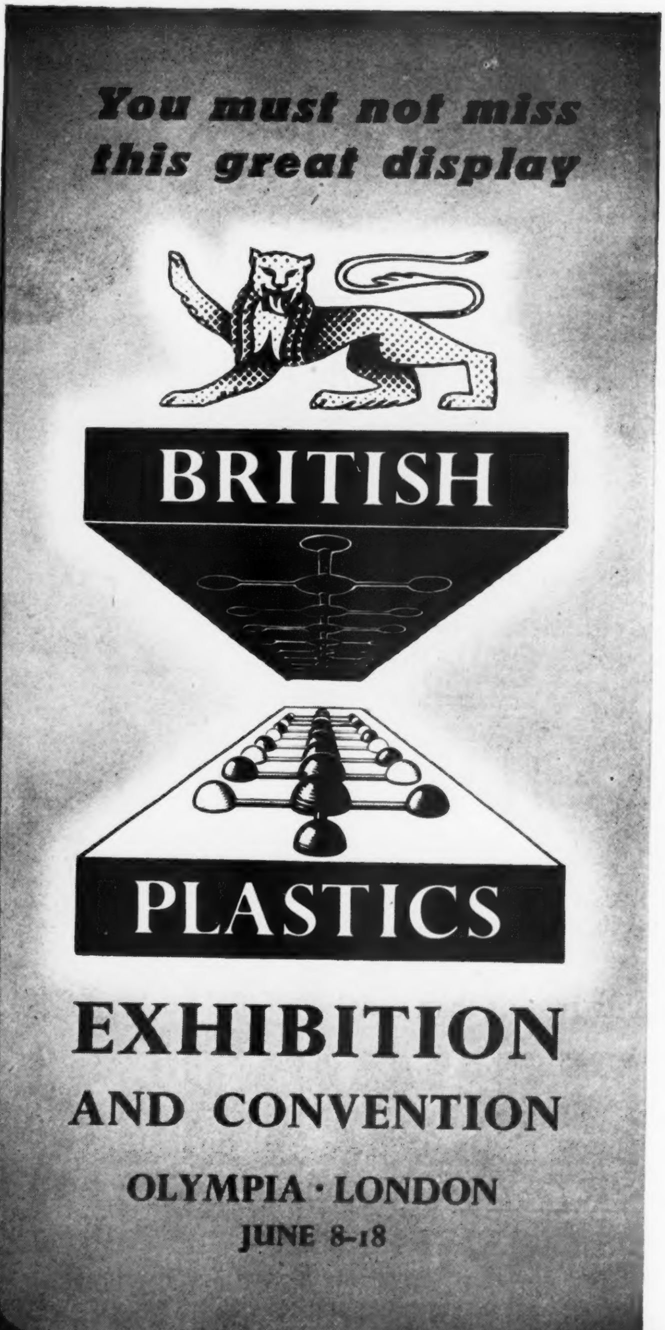

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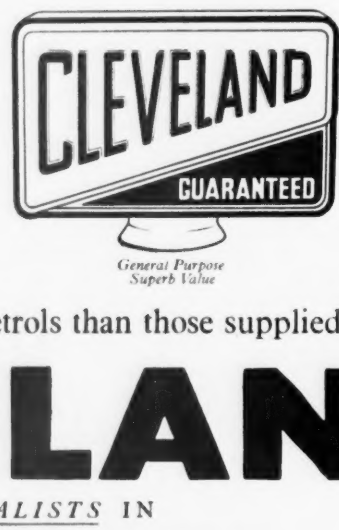

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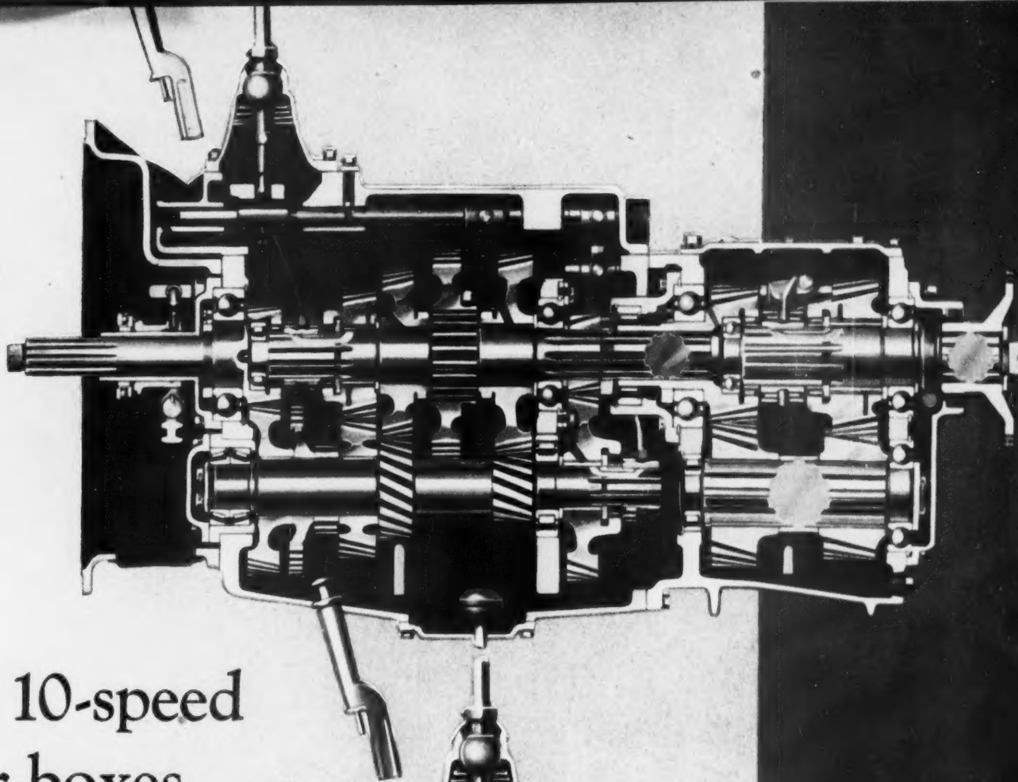
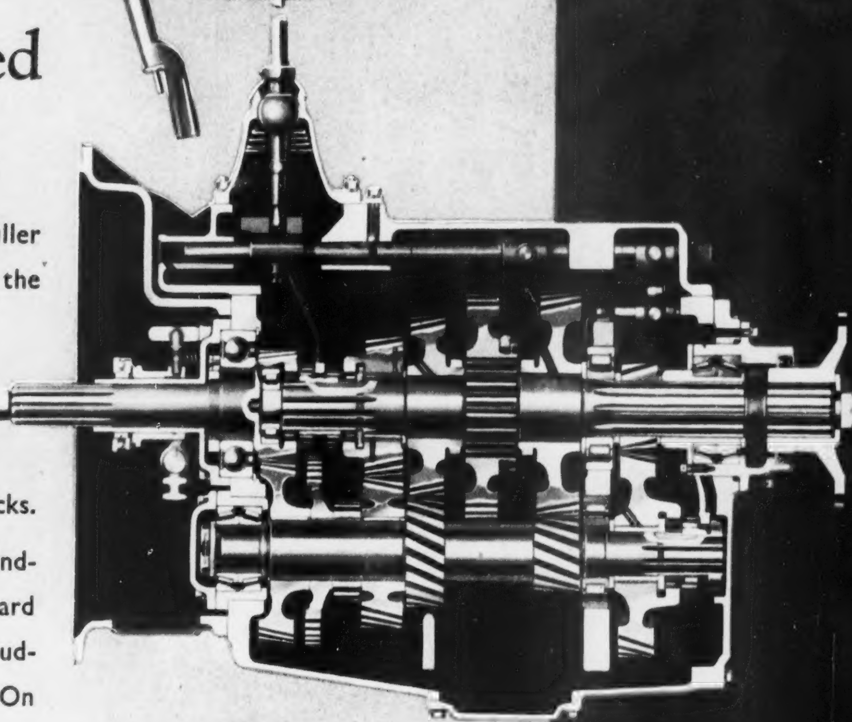


5 and 10-speed gear-boxes



The top illustration shows the Fuller 10-speed gear-box, comprising the famous five-speed box with a Fuller two-speed auxiliary box built on to it, thus providing a ten-speed box for heavy-duty operation on large trucks.

This unit is to the usual Fuller standard of high-duty, with all forward gears helical, and all changes, including reverse, by dog-clutches. On both of these boxes the gears are shot-peened and crown-shaved, to avoid stress concentration.

The lower illustration shows the Fuller 5-speed gear-box. Every gear is helical and engaged by dog-clutches and to reduce shaft deflection to a minimum the mainshaft is supported on three bearings, and the layshaft kept short, making one of the highest-duty gear-boxes ever produced.

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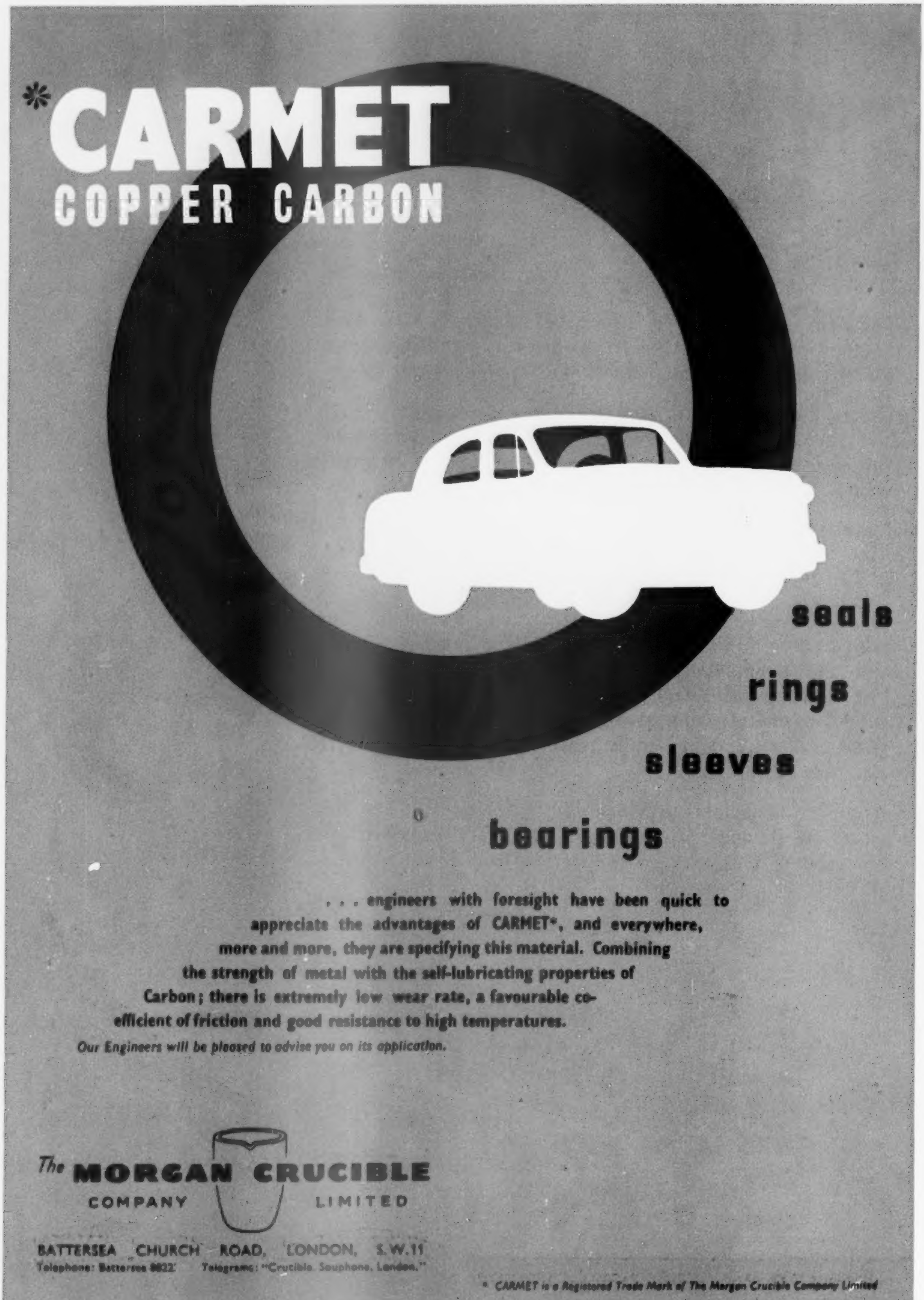

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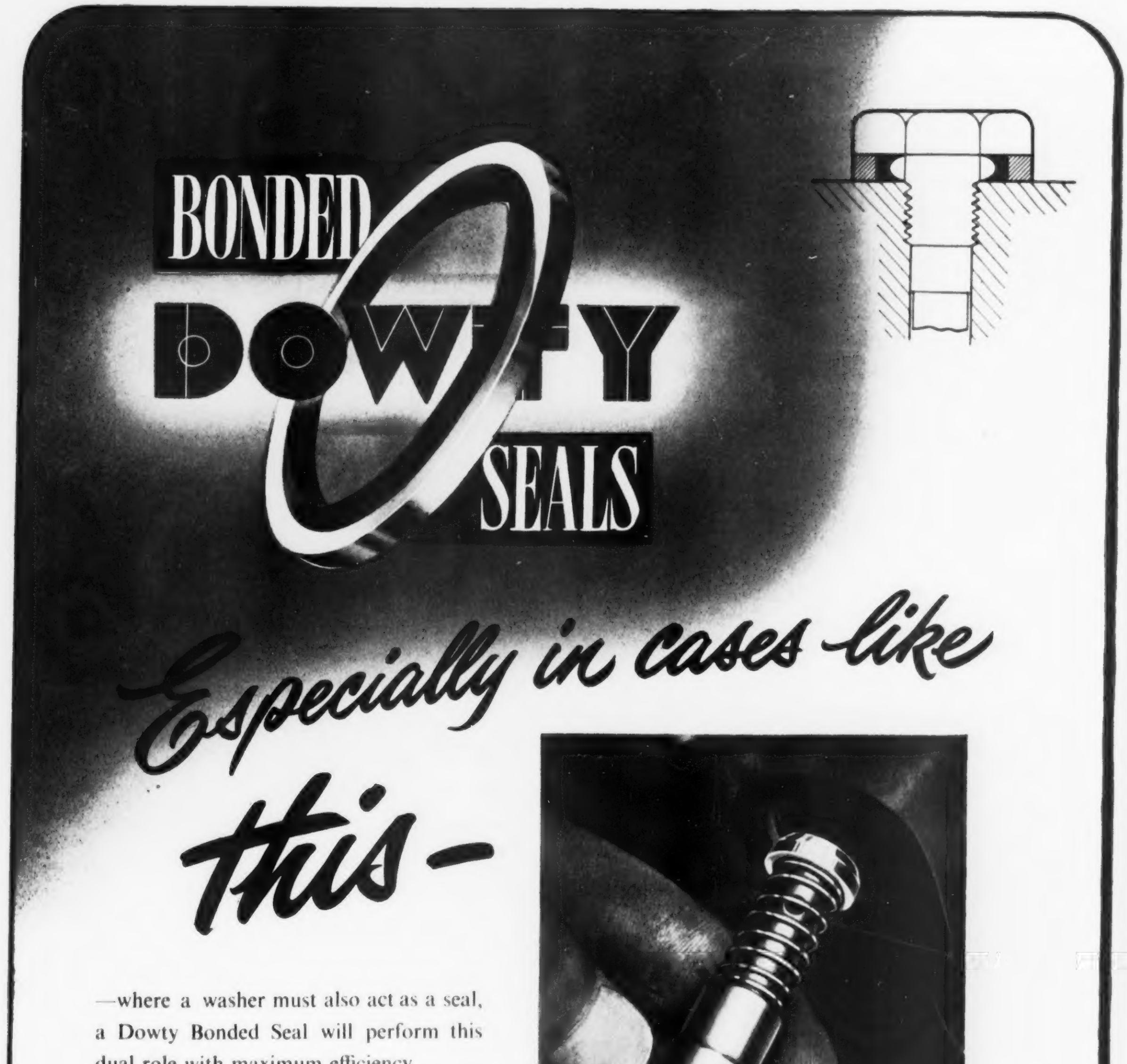
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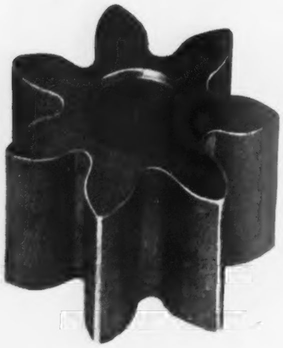

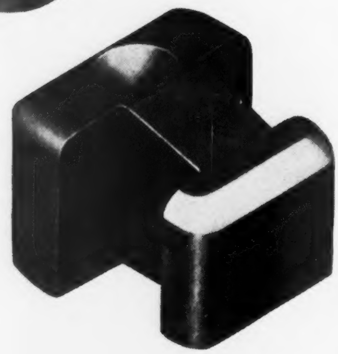

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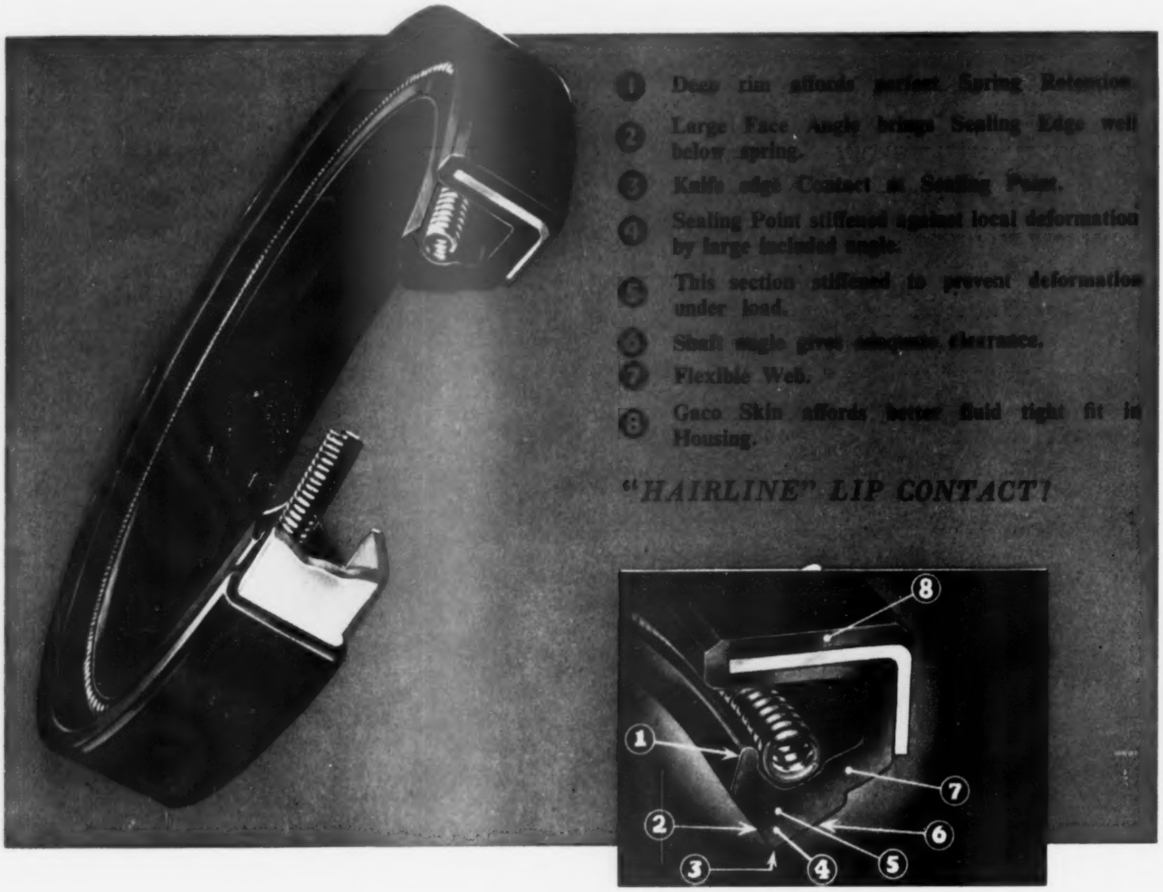

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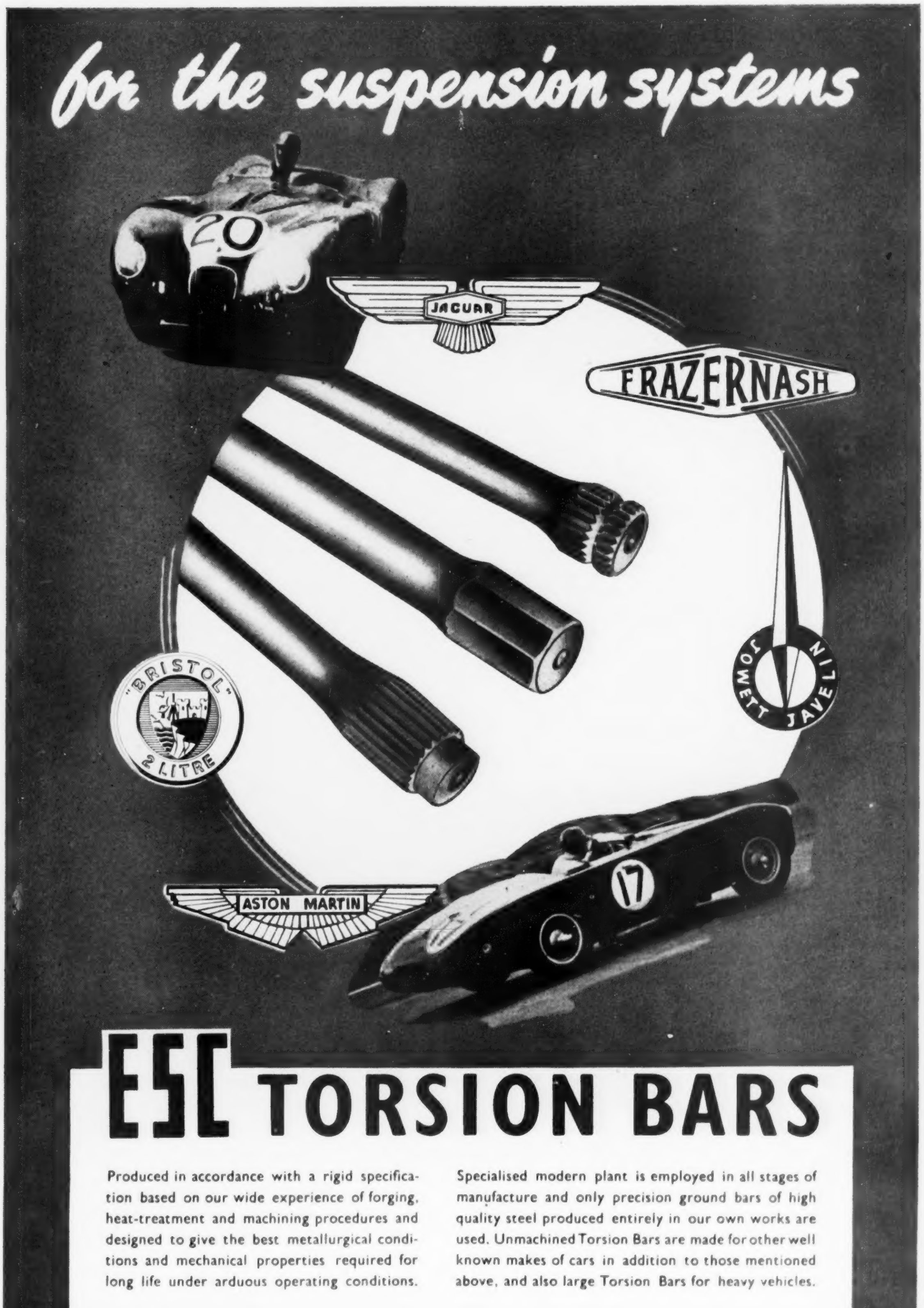
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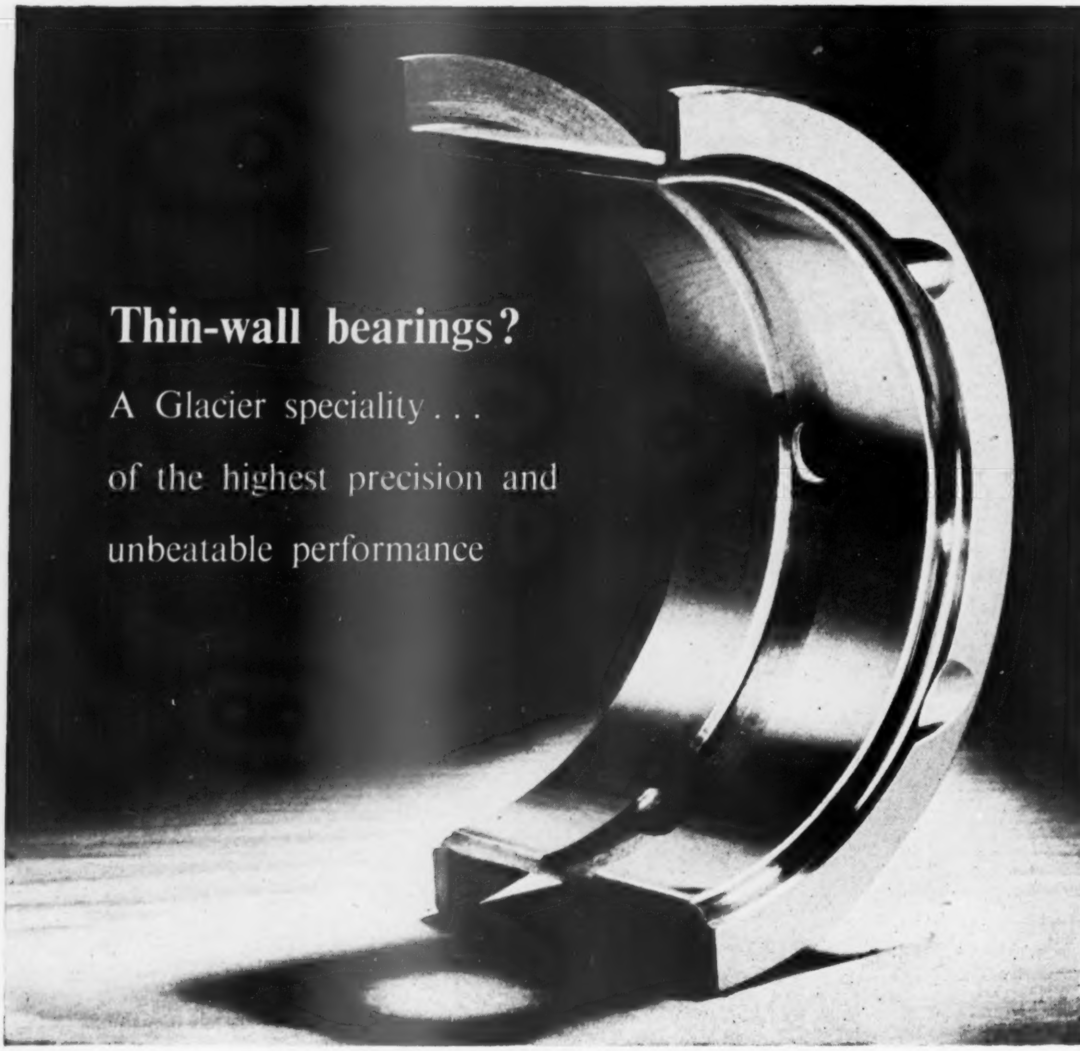
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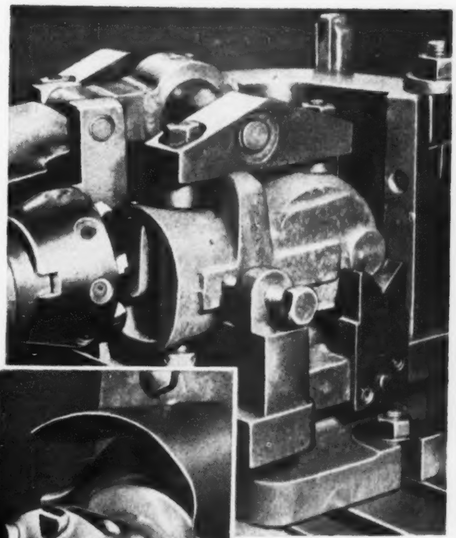

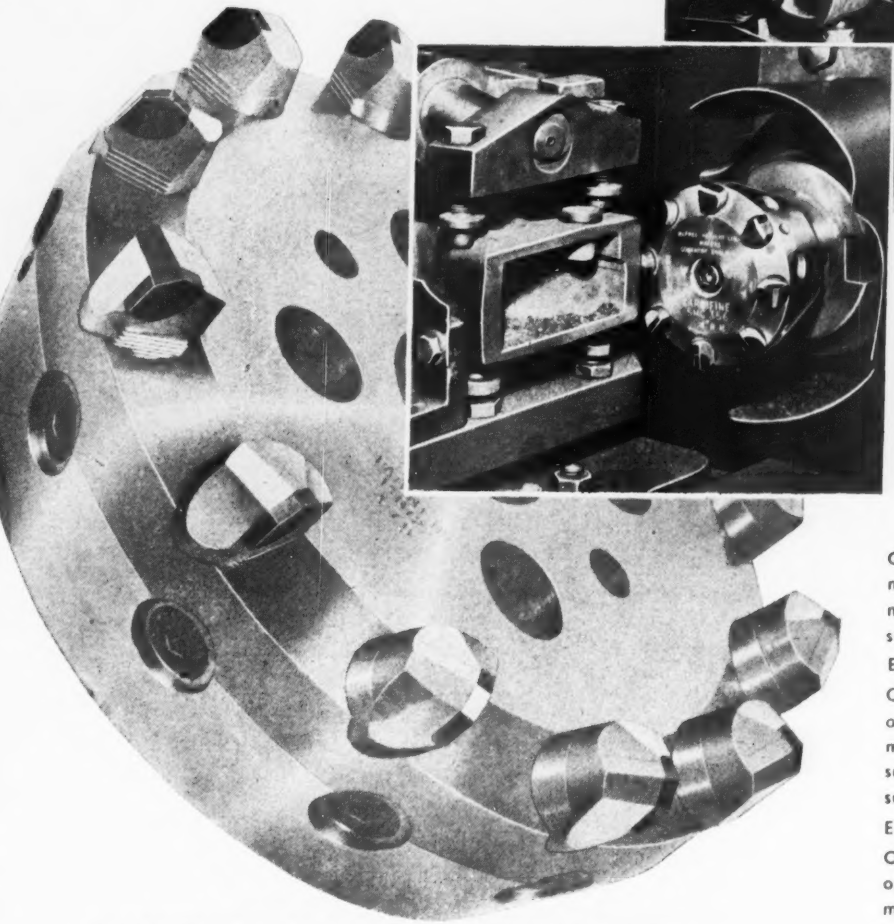

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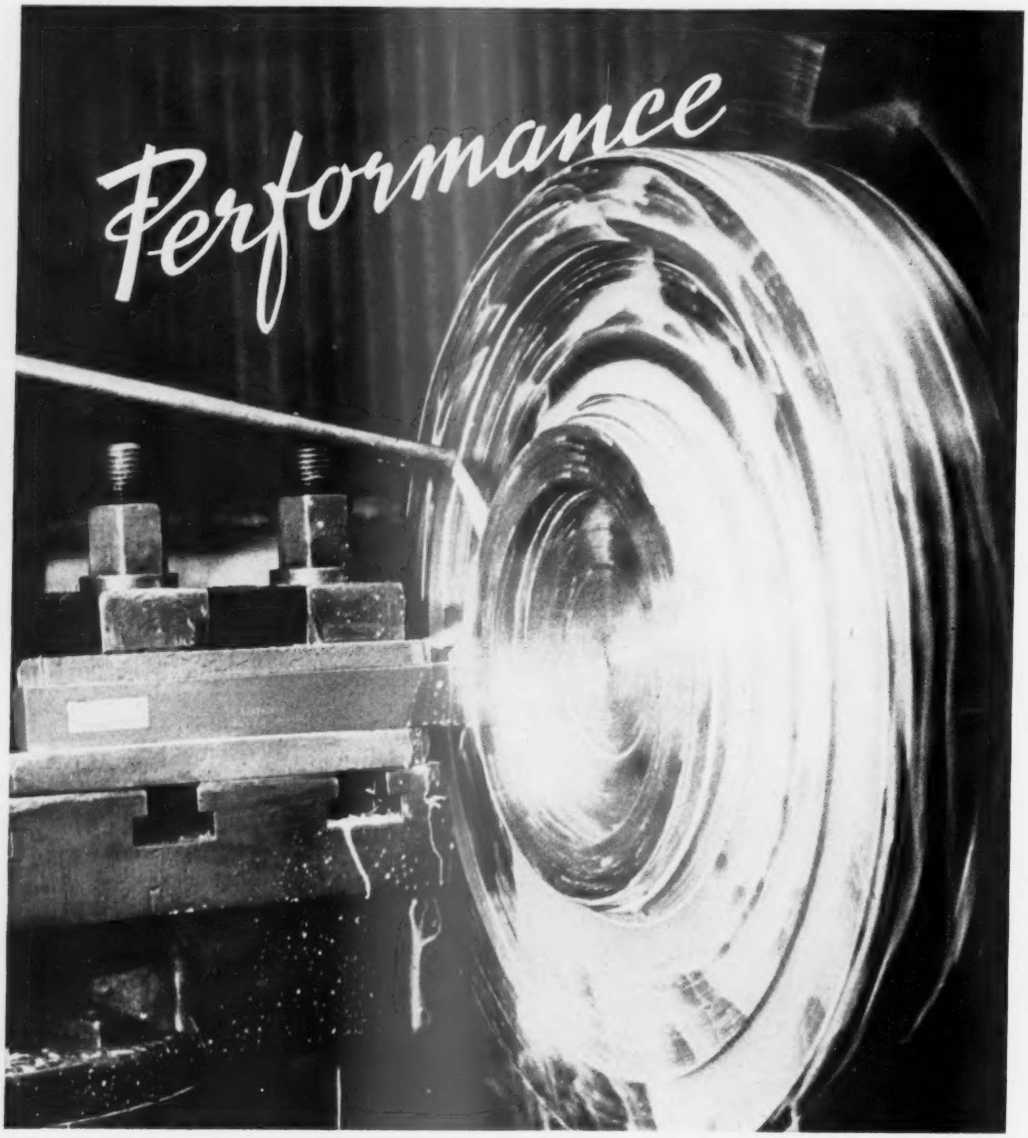
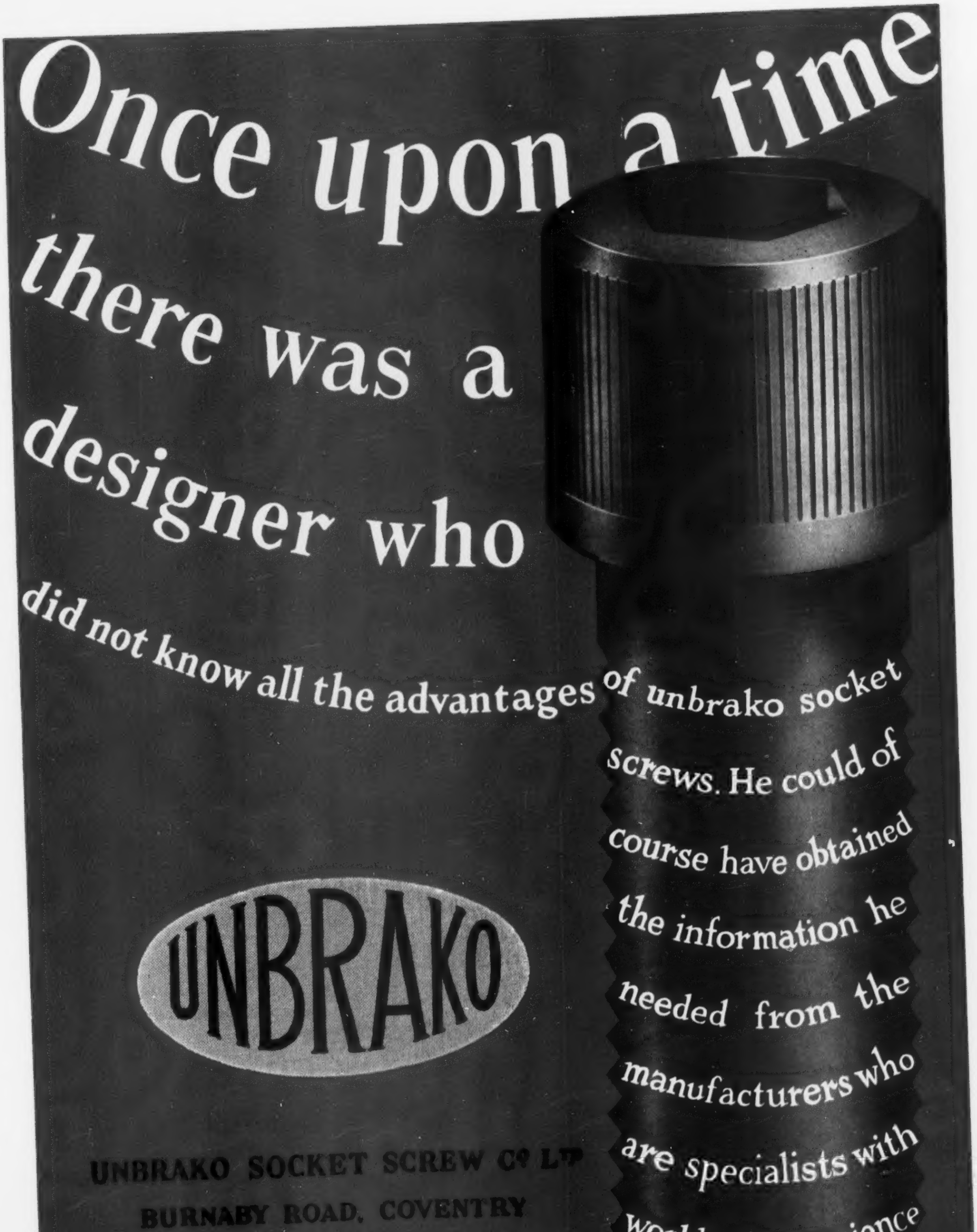

Performance


Once upon a time
there was a
designer who
did not know all the advantages of unbrako socket
screws. He could of
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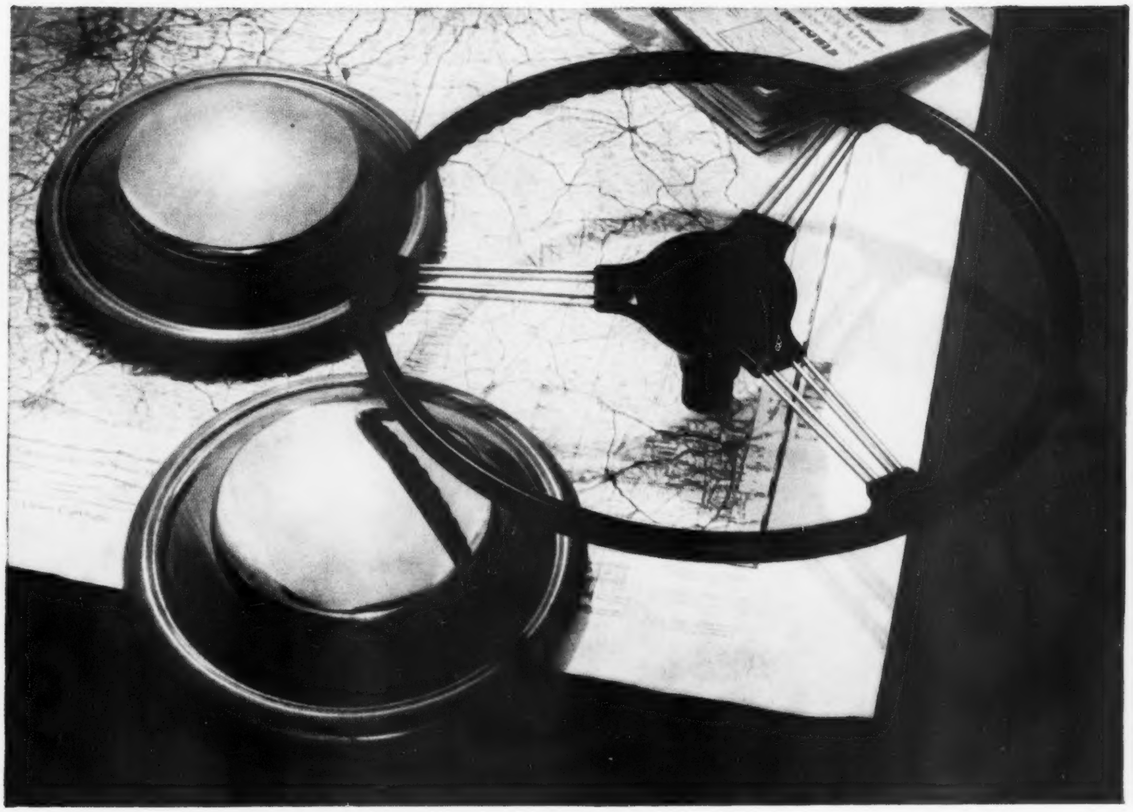

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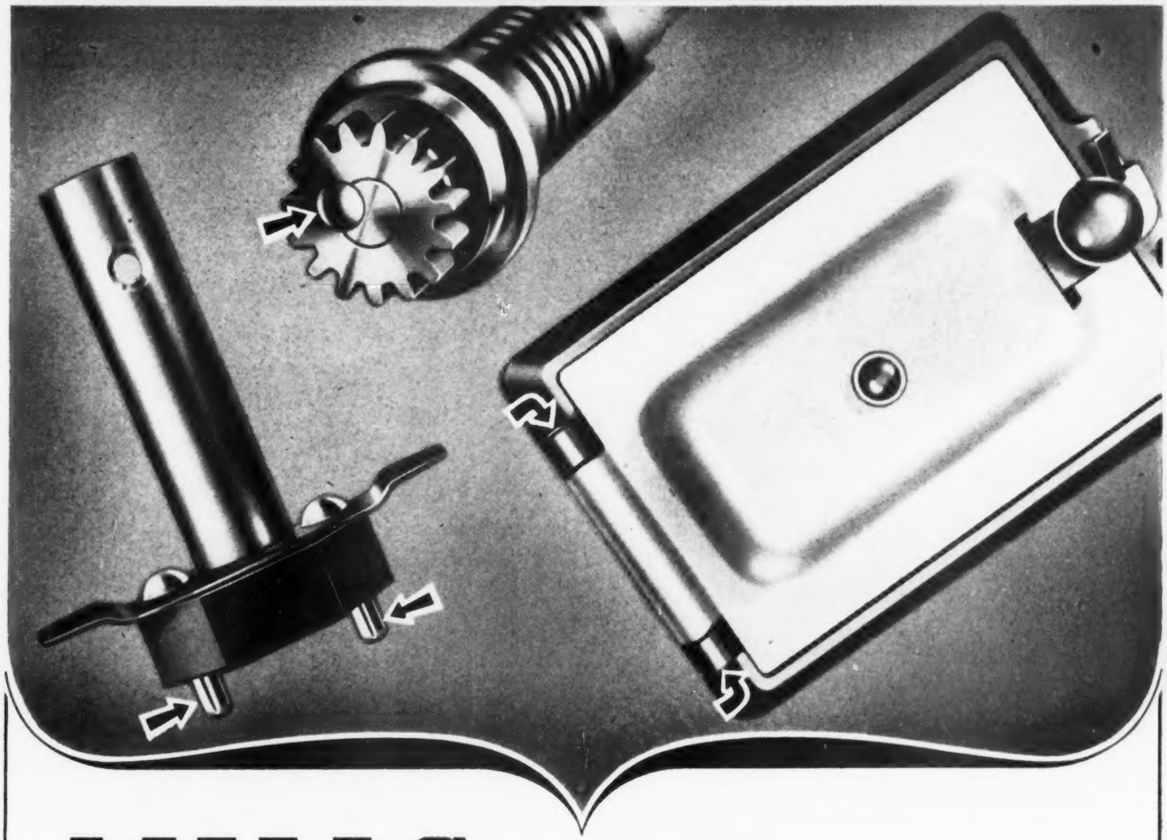

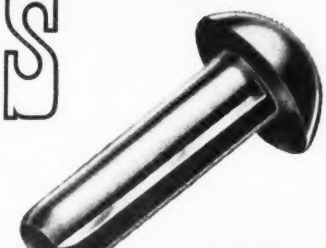

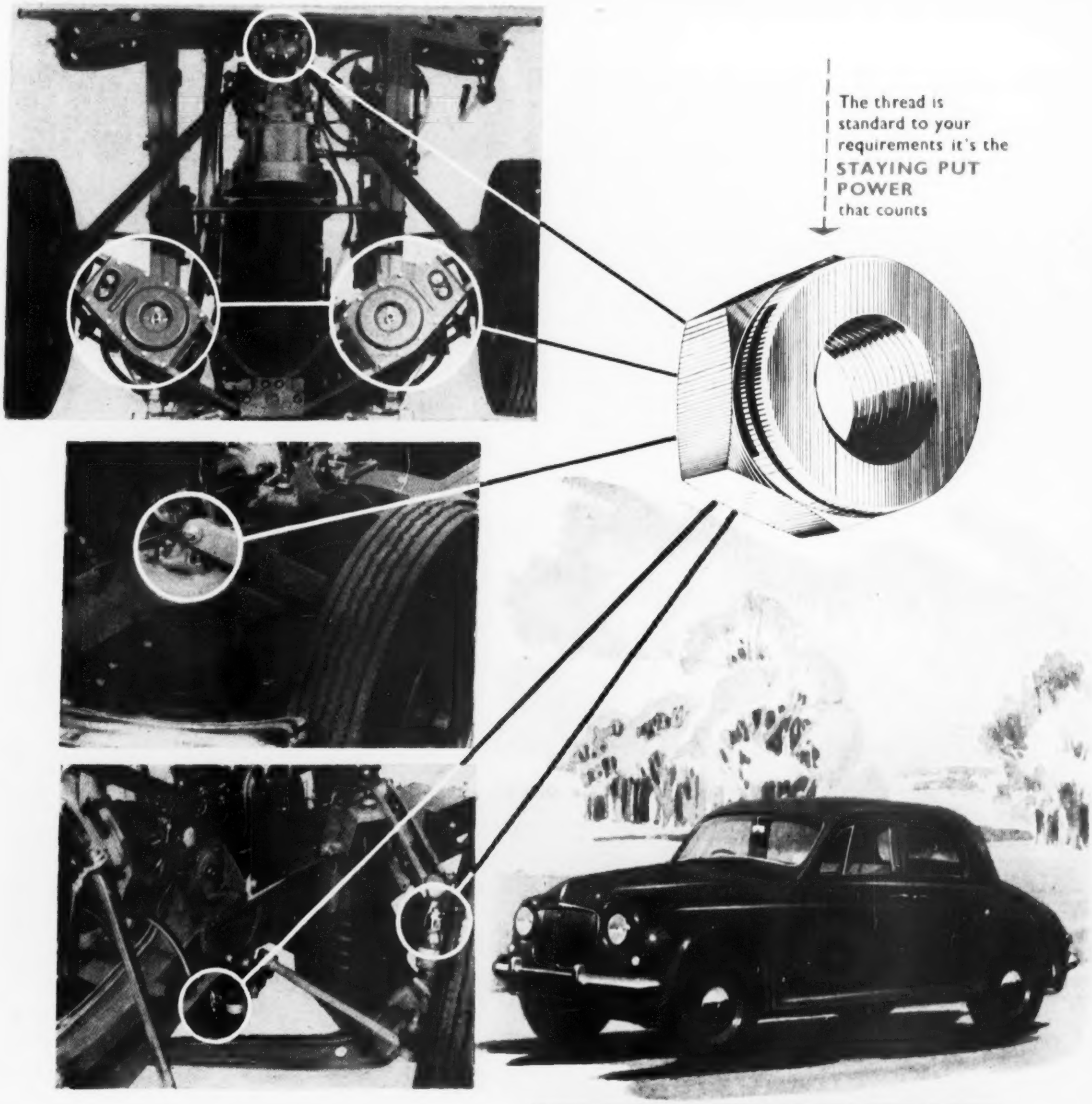

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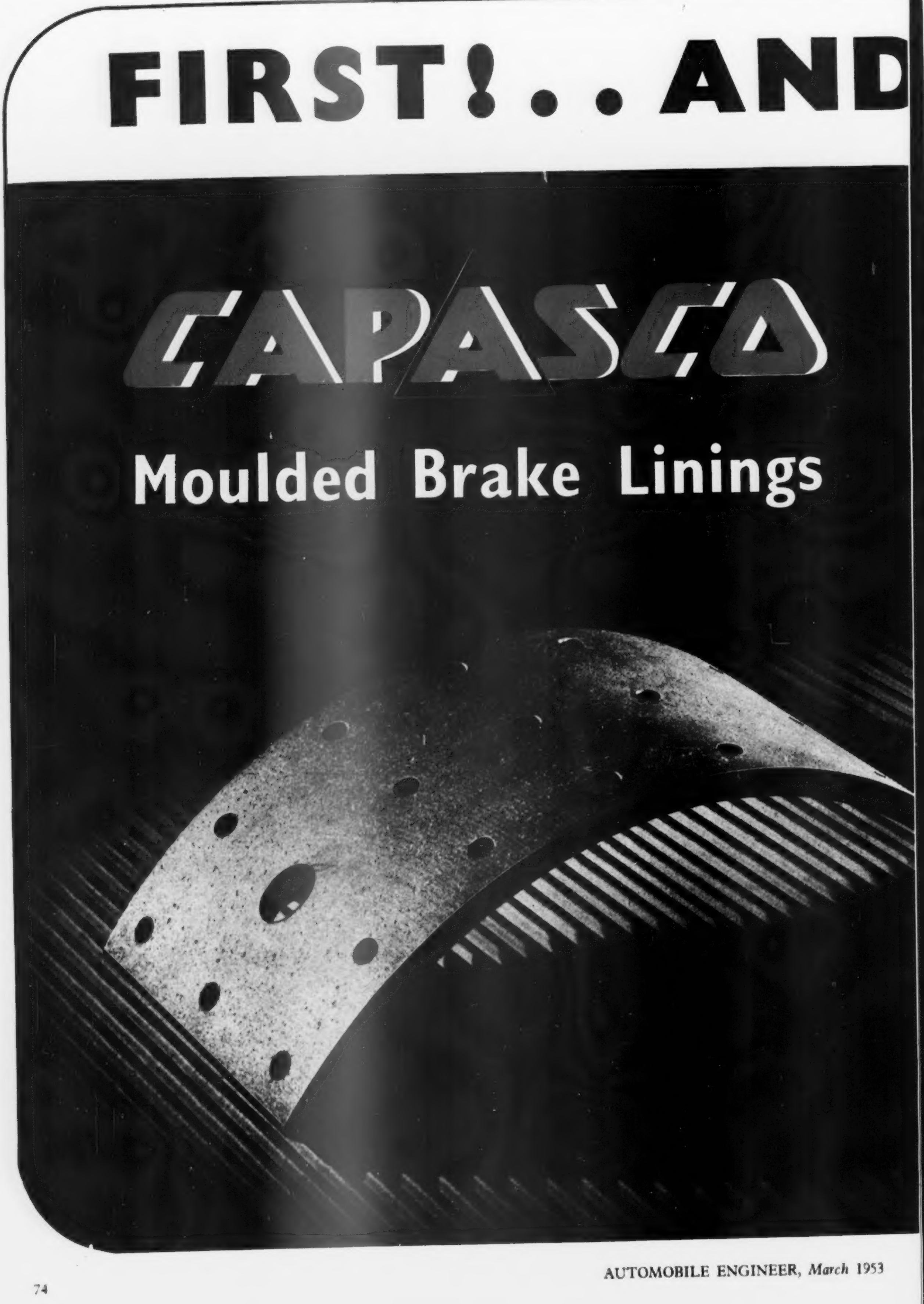
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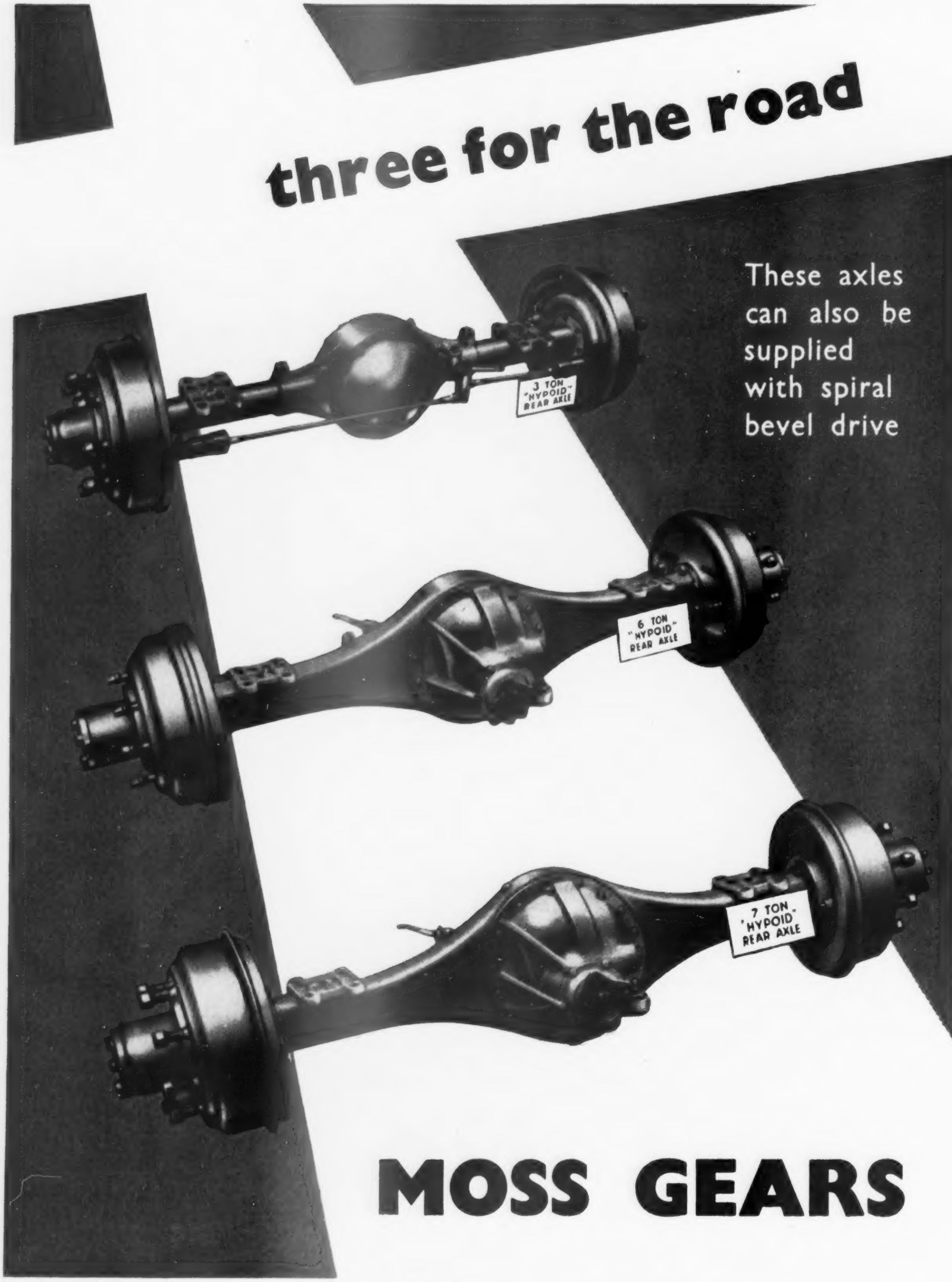
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Design, Materials, Production Methods, and Works Equipment

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Power Output

CLAIMS that have recently been made concerning the increased b.h.p./capacity ratio for certain American automobile engines, have led to our carrying out an analysis for most of the current British engines and for one or two American and Continental designs. The results were such as to cause some little surprise, and on the face of things they were in some respects rather disturbing. That there should be an appreciable divergence in the ratios for an engine for a low-price vehicle and an engine for a fairly high-price vehicle was only to be expected, but it was disconcerting, to say the least, to find that of two engines for vehicles in the same price range, one claimed a 90 per cent greater maximum power output per cubic inch of swept volume than the other.

It is pleasing to know that of all the engines considered, an English design had the highest ratio. It was substantially higher than any figure yet quoted for the latest designs of American engines. To engineers conversant with the engine in question the Jaguar Mark VII, this will not be surprising. It is well known that in the development of this engine, considerable experimentation was carried out to determine the shape of combustion chamber and the engine breathing characteristics generally needed to give the optimum results in service.

The importance of combustion chamber shape and breathing characteristics are recognized by all automobile engineers, but only too often they receive only lip service. This is to be regretted, since careful attention to these details undoubtedly offers the cheapest method of obtaining greater output from a given capacity. The development work would, of course, take longer and be more costly, but once the optimum conditions were established, production costs should not be materially affected. Any extra development cost would be amortized over a relatively large number of engines and the added cost per unit would be small.

That the American manufacturers are striving to obtain substantially increased output per cubic inch of swept volume and are publicizing the results, suggests that they may once again soon be striving to regain the position they held in export markets in pre-war years. For the time being, the publicity seems to be aimed at the American home market, but it would be foolish to discount the possibility that it may soon be aimed at a wider market.

Inevitably, many factors must influence the share of export markets that will fall to individual manufacturers and individual countries, but the efficiency of the power unit will always remain one of the most important. It is therefore, desirable that steps be taken to ensure that British power units are developed to give optimum efficiency. Although the b.h.p./capacity ratio is not an absolute measure of efficiency, it will generally be found that an engine with a high ratio has a higher overall efficiency than a lower ratio engine.

A high ratio is definite evidence of good combustion and breathing characteristics. It is therefore practically certain that the rate of carbon deposition will be appreciably lowered. There is a two-fold advantage from lower carbon deposits. First, the engine remains at near its original efficiency for longer periods and therefore will have better fuel consumption figures; and second, the intervals between decarbonizations will be greatly extended.

Materials Handling

IN the past four or five years the automobile industry has shown a very much keener interest in materials handling than ever before. Even in pre-war years, the industry showed much more interest than any other engineering industry in the problems concerned with the movement of materials. Then, however, the interest was almost solely concerned with movement within the factory, and substantial economies were effected by using mechanical conveyors to transfer work from one location to another. These developments gave savings in direct labour costs, and also gave less obvious savings in that the conveyors acted as stores, and so released for productive purposes floor areas that would otherwise have been needed as storage space.

During the post-war years there have been further developments in handling materials within the factory. Of these, probably the most important is the wide use now made of fork-lift trucks. These have also reduced the superficial area needed for stores, since they have made it easy to store vertically as well as horizontally.

It is substantially correct to say that until comparatively recently, the problems of materials handling were considered as being confined within the four walls of the factory. In fact, that is still the attitude in some organizations. However, an increasing number of planning engineers now realize that for maximum efficiency, the control of

materials handling must start at the despatch bays of suppliers' factories.

The first step in this direction was taken when certain component suppliers were persuaded to despatch their products in palletized loads of uniform quantities. This practice again reduced the need for storage areas, since the palletized loads could in many cases be transferred direct from the receiving bay to the production department. Unfortunately, even now the best possible use is not made of this practice ; only too often the supplier uses a type of pallet that is far from satisfactory to the user of the product. This is a matter that calls for the closest co-operation between the supplying organization and purchasing organization. Undoubtedly, the return of the buyer's market will lead to an improvement in this respect.

A more recent development that, so far as we are aware, is peculiar to the Austin Motor Co. Ltd. has great potentialities in the matter of transport between supplier and user. Briefly, the Austin Motor Co. Ltd. ask that in suitable cases standard trailers be used for delivering materials to their factory. This system is a logical development of the unit pallet load system. It is already working successfully in the case of radiator supplies.

The Austin Motor Co. Ltd. have gone very far in reducing the volume of materials kept in stock and they hope to go even farther. The "trailerized" loads (we apologize for the barbaric term that has been adopted as a designation for the system) should do much to help towards further reduction of stocks. As the radiators are finished, they are loaded into a trailer that gives them complete protection against the weather. The trailer stands out in the open until its load is called for. The adoption of this system has released for productive use in the radiator factory, an area that would otherwise have been required for stores.

When further supplies are needed at the Austin factory, an empty trailer is sent to the radiator factory and a full one is brought back. The full trailer stands outside at the Austin works until its contents are required. This system has substantial advantages, but it does call for close co-operation between the supplier and the automobile manufacture if the full benefits are to be obtained. For maximum efficiency and economy, supplies should be received in low volume at high frequency. Only so can the capital locked up in work in progress be kept to a minimum, an important factor at any time, but now of more than usual importance because of our present fiscal and monetary policy.

This system greatly simplifies the problem of storage space. All that is required is a concrete floor on which the trailer can stand until it is unloaded. There is therefore a much lower capital charge than would be needed for a conventional stores. All-in-all, there is little doubt that properly applied this system can show economies for both the supplier and the user.

American Trends

INFORMATION recently received from an engineer associated with American automobile manufacture suggests that the trend of developments in the United States will most probably follow a pattern very different from what has been expected by automobile engineers in this country. The cardinal point that our informant made is that basic price is the decisive factor in sales. So much so in fact that one model had bad sales last year merely because the basic price of an immediately competitive model was 50 dollars lower.

We in this country have become accustomed to thinking that American trends were towards greater and greater complexity, as indeed they have been. Apparently a reversal of policy is now imminent. Whether the change in policy will be effective remains to be seen. Many American car owners have already become conditioned to simplified driving through the use of automatic transmissions ; will they be prepared to return to the manual use of the gearbox?

If our informant is correct in his suppositions, automatic transmissions as we now know them and power assisted steering will be fitted to many fewer cars, but the demand for them has been created and will continue. No doubt the American car owner is like his British counterpart. He wants a low basic price but he will still expect the refinements to which he has become accustomed. Designing to a price and still attempting to meet all the wishes of prospective owners is a thankless task. It will be interesting to learn how American designers meet it.

It may be taken for granted that, in the U.S.A. at least, automatic transmissions have come to stay, but not necessarily in their present form. The probability is that there will be intensified efforts to develop a much cheaper and simpler form of partly automatic transmission. This seems to be the logical development for the American home market. The development of such transmissions will be easier in America than in this country. Here, fuel costs are important, in the U.S.A. they have little or no effect on the choice between different makes of cars.

CONTENTS

ENGINE MOUNTING

Fundamental Principles and Mounting Details

THERE are many who do not fully understand the principles that must be applied to the design of engine mounting systems for road vehicles. This is not surprising since few, if any, modern text books deal with the subject at all comprehensively, and the majority of papers cover only specialized sections of the problem. As a result, there is in some quarters a lot of muddled thinking, and many fallacious theories still remain to be discredited.

Before motor driven, self propelled vehicles were manufactured on a commercial scale, the general practice as regards the mounting of stationary engines was to anchor them as securely as practicable to a firm foundation. It was natural, therefore, that the same practice should be continued in the motor vehicle, and that the foundation on which the engine was mounted should be the chassis frame.

However, it is of interest to note that in many cases the frame was not a stiff enough foundation, and that because of weaving and other forms of distortion, it was often found necessary to isolate the engine from the effects of frame movement. This was done either by mounting the engine on a sub-frame which usually had a three-point attachment to the main frame, or by incorporating some other device to prevent loads from being transmitted through the engine crankcase.

Although it is more usual nowadays to think of flexible engine mountings as a means of isolating the frame from the vibrations of the power unit, the converse must also be considered. This is particularly so for commercial vehicles in which frame distortion may be considerable and may impose severe strains on rubber mountings. Some manufacturers have even gone to the opposite extreme, and use the crankcase as a structural member in the frame. This practice is currently adopted in vehicles such as tractors, for which small size or extreme sturdiness is more important than freedom from transmitted engine vibration. Indeed, the crankcase and transmission casings in many cases perform the functions of the chassis frame.

Two main considerations make it desirable to isolate both commercial vehicle and private car structures from engine vibrations. One is that noise and transmitted vibration are fatiguing. Therefore, a vehicle with a well mounted engine is not only more pleasant to drive than one with a poorly designed system, but also accidents due to driver fatigue are less likely. The other consideration is that engine vibrations apply alternating loads to the vehicle structure, which may cause fatigue failures. These failures may be costly to repair and are liable to cause loss of goodwill towards the manufacturer. A third factor, of importance particularly where private cars are concerned, is that complete freedom from vibration and noise is associated by the general public with the more expensive luxurious cars, and is a valuable selling point.

Fundamental requirements

The primary object in supporting a power unit on flexible mountings is to isolate the vehicle structure from engine vibrations. In order to do this as effectively as possible the natural frequency of the engine on its mountings must be as low as practicable, relative to the forcing frequency, for all the appropriate modes of vibration. This follows from the fact that the proportion of vibration transmitted, or the transmissivity, is given by:

$$T = \frac{1}{\left(\frac{f}{f_n}\right)^2 - 1}$$

where f = the forcing frequency, and

f_n = the natural frequency of the system.

Engine designers are, of course, familiar with the vibrational characteristics of various types of engine, which may be recapitulated briefly. There are primary out-of-balance forces due to the main reciprocating masses comprising the pistons and the proportion of their connecting rod masses, and the secondary component which is due to the modification of the simple harmonic motion by the connecting rod. There are higher harmonics but their amplitudes are small and frequencies so high that they are rarely of any practical significance. In addition there are torsional vibrations due to the firing impulses and to the inertia of the reciprocating masses. Forces due to incompletely balanced rotating parts may also be experienced in a few designs.

Whether or not the other vibrational characteristics mentioned can be eliminated depends on the engine layout. For instance, in a four cylinder in-line engine, the primary forces are eliminated since, with the plane crank arrangement, the pistons move up and down in pairs, and the motion and hence the forces due to one pair are opposite to and balance those due to the motion of the other. Moreover, since the pairs are each symmetrical about the centre of the block, there are no out-of-balance couples. However, there are secondary out-of-balance forces since the pairs of cranks are spaced 180 deg apart, and the secondary frequency is twice the primary. It follows that when the cranks have turned through 180 deg the secondary has moved 360 deg and the forces are therefore added. In a six cylinder engine with cranks at 120 deg, all inertia forces of any practical significance balance out; moreover the torque also is relatively smooth.

The frequency that is often of particular importance so far as engine mountings are concerned is the half order frequency in four stroke engines and the first order in two stroke units. This is the frequency of vibration due to uneven strength of firing impulses, caused by imperfect mixture distribution to the cylinders. It is of importance for two reasons. The first is that most engines have this defect at least in a small degree, and the second is that it is a relatively low frequency vibration. The 1, 1½, 2, 2½, 3 and 3½ etc. orders may occur when the firing impulses of the different cylinders are unequal. Other vibrations may be excited by the valve gear and camshaft. However, these are generally of small magnitude, and are usually automatically catered for by a mounting designed to absorb the half order already mentioned.

The lowest frequency of engine vibration that must be isolated from the vehicle structure determines the degree of flexibility necessary in the mounting. In fact, for satisfactory isolation the natural frequency of the power unit on its mountings should be no more than half that of the exciting frequency. The natural frequency in torsion of a flexibly mounted engine is given by $f_t = 9{\cdot}55\sqrt{\frac{Q}{J}}$ cycles/min, where Q = the stiffness of the mounting system in lb-in/radian, and J = the mass moment of inertia, in lb-in^2 about the principal axis. So far as vertical vibrations are concerned the natural frequency is given by $f_n = \frac{188}{\sqrt{\delta}}$ cycles/min, where δ = the static deflection, in inches, of the mounting under the weight of the power unit.

There is another factor, besides natural frequency, which must be considered in design. That is the size of mounting which must be employed. The size is determined by the strength of mounting necessary to support the weight of the engine under dynamic loading due to the motion of the vehicle when travelling over rough terrain. Furthermore, bottom gear torque reaction must be catered for, as well as the largest amplitudes of vibration, due to internal out-of-balance forces, of the engine on its mountings.

Motion of the vehicle over rough terrain produces maximum vertical acceleration of 3 g according to Johnson and Heyl[1]. Experimental work carried out in this country seems to indicate that rebound loads are in some cases approximately equal to bump loads, although the reason for this is somewhat obscure. It is considered advisable

therefore to consider the 3 g load as being reversible. Loads of ± 1 g may be experienced longitudinally due to braking and acceleration, and ± 1 g laterally due to cornering. A factor of safety of 1·5 should be applied to these loads. It would appear likely that a factor of 2·0 on the maximum engine torque will cater for bottom gear conditions when the clutch is let in suddenly.

The amplitudes of motion due to the vertical or, where appropriate, the horizontal out-of-balance forces may be determined from the equation:

$$m\frac{d^2x}{dt^2} + kx = P_o \sin \omega t$$

where m = the mass supported on the engine mountings, x is the deflection from the static position, k = the stiffness of the mounting system, and $P_o \sin \omega t$ is the out-of-balance force. The solution of the equation is given by Den Hartog[2] as:

$$\frac{x}{x_{st}} = \frac{1}{1 - (\omega/\omega_n)^2}\sin \omega t$$

where x_{st} = the static deflection of the engine on its mounting, ω_n is the natural frequency of the system, and ω is the forcing frequency.

Torsional amplitudes may be derived from the equation

$$\frac{Jd^2\psi}{dt^2} + Q\psi = T_o \sin \omega t,$$

where J is the mass moment of inertia about the appropriate principal axis, ψ is the amplitude of angular motion, Q is the torsional stiffness of the mounting system, and $T_o \sin \omega t$ is the torsional out-of-balance force. The solution is similar to that for the translatory forces. A magnification factor $\frac{1}{1 - (\omega/\omega_n)^2}$ is obtained, and multiplied by the deflection that would occur if the out-of-balance torque were applied statically to the mounting system. The resulting deflection must then be added to that due to the appropriate steady torque, and the total will be the maximum deflection under normal running conditions. This will rarely exceed the torque due to suddenly engaging the clutch in bottom gear.

Another torsional vibration which must then be considered is that caused by rocking couples. These may be caused not only by reciprocating unbalanced forces acting about a point on the transverse plane of symmetry of the cylinders, but also by the same forces acting about the centre of gravity of the engine and gearbox unit. This follows from the fact that the centre of gravity is rarely on the plane of symmetry of the cylinders, and forces passing through any point other than the centre of gravity will cause rotation of the unit as well as translational motion. The principles already described again apply for determining what must be the natural frequency in torsion in this mode, as well as the amplitudes of motion. However, in conventional four cylinder engines, the only rocking couples normally experienced are those due to the secondary unbalance forces acting in the plane symmetry which is offset from the centre of gravity. Six cylinder engines with a 120 deg crank arrangement may be completely balanced, so there are no rocking couples.[5]

Fig. 1. Centrifugal forces acting on a spinning body tend to make it rotate about its principal axis

At present very little information is available about the magnitudes and modes of chassis frame deflections. Undoubtedly it is a much more serious problem in commercial vehicles than in private cars. However, it is felt that experimental work on the subject, if methodically analyzed, would yield results which could be translated into empirical data that could be used in engine mounting design, as well as in other connections.

Such experimental results would have to be related to different forms of structure and cross sections, various widths and lengths of frame, as well as weights of vehicle, and the position and number of body mounting points. Thus it can be seen that to deal with the subject adequately, the experimental programme would be a long one, and the results would probably give only approximate indications as to the deflections likely to be experienced.

The physical properties of rubber mixes vary so widely that it is only possible to give some approximate rules for guidance in determining the allowable stresses.[4] In compression the static strain should not exceed 10-15 per cent and the allowable strain varies with the area ratio. This in bonded units is the ratio between the area of rubber bonded to metal and the free area of rubber. When the area ratio is high, that is when the rubber sandwich is thin by comparison with is bonded area, the lower limit will be suitable, but when it is low the upper value can be used.

Under shear, the allowable static stress is not so high, and should be limited to 50 lb/in², or to such a value that the strain does not exceed 80 per cent. The stress appropriate to the lower of these two conditions should be adapted for design purposes since, with soft grades of rubber, the stress at 80 per cent strain will be below 50 lb/in². Rubber is much more flexible in shear than in compression, and advantage of this property may be taken in the design of mountings.

It is not advisable to load rubber in tension. This is because under these conditions minute crevices tend to open and draw in corrosive substances, principally ozone, which shorten the life of the unit. In mountings loaded in shear, there is also an associated principal tensile stress, but if a compressive pre-load is applied to eliminate the tensile component, the life of the unit may be increased. All the design criteria given in this and the preceding two paragraphs are of general application. However, for engine mountings, a useful rule of thumb is to limit the shear stress to 30 lb/in² and the compressive strain to 12 per cent under the dead weight of the engine and with maximum bottom gear torque applied.

Arrangement of the system

Two main considerations influence the layout of the mounting system. One is that the out-of-balance forces due to the reciprocating masses act in the transverse plane of symmetry of the cylinders. The other is that torsional vibrations take place about an axis which passes through the centre of gravity of the engine and gearbox unit, and which is rarely either horizontal or coincident with the axis of the crankshaft journal bearings.

The first stage in the design process is to estimate the position of both the centre of gravity and the axis about which the unit will vibrate torsionally. At the drawing board stage this can only be guesswork based on experience of previous engine designs. For this reason it is essential that the supports for the rubber mountings should be designed in such a way that their position may be altered during the development stage without involving costly tooling changes. This can be arranged in several ways, one of which is to make use of a flange on the engine plate to carry the front mounting, and to fit suitably designed bolted-on brackets at the rear. If the best possible results are to be obtained to give complete freedom from engine excited vibration and noise in the car, the mounting design cannot be finalized until a prototype has been made. Only then can the necessary data be obtained experimentally.

First the location of the centre of gravity must be

determined exactly. For this operation all the engine accessories such as air cleaners, etc., must be on the engine. A mass equal to ½ of the weight of the propeller shaft should be bolted to the companion flange for the universal joint at the rear end of the gearbox mainshaft. The centre of gravity of this mass should be at the position of the universal joint centre. A mass equivalent to the oil must be wedged in the sump. This mass should be of a material of roughly the same specific gravity as oil in order that it may be cut to approximately the same shape and size as that portion of the sump normally occupied by oil, and so that the centre of gravity position will be the same. No doubt the error would be very small if paraffin wax were run into the sump and allowed to set.

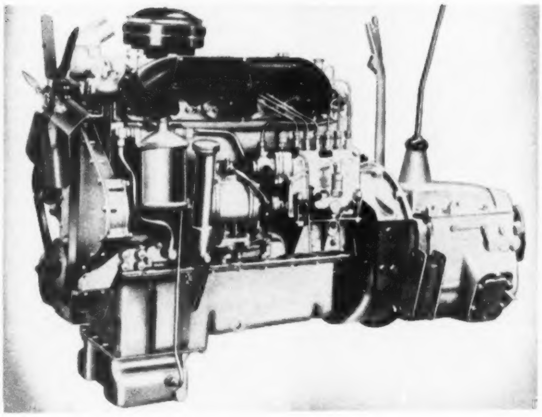

Fig. 2. The Perkins P6 engine mounting installation in the Dodge truck

The centre of gravity check may be made in a number of different ways, but a convenient method is as follows. A beam is supported horizontally beneath two spring balances, one at each end, arranged so that they each carry an equal proportion of the weight of the beam. The supports for the spring balances should be such that their height may be adjusted to maintain the beam in a horizontal position when it is loaded. A centre line is marked transversely exactly mid-way between the two supports. Next, saddles are mounted over the beam to carry the engine slings. The slings should be of such a length that they will support the engine below the beam with its cylinder axes parallel to the vertical plane containing the longitudinal axis of the beam. A turnbuckle may be incorporated in the sling to facilitate final adjustment. Machined faces on the cylinder block or crankcase are suitable for use at datum planes.

Next, suspend the saddles and slings from the beam in the appropriate position relative to one another to support the engine in the correct attitude. They should also be so disposed about the centre line that each balance supports an equal portion of the total weight. Remove the slings, and attach them to the engine which should then be lifted beneath the beam so that the slings may be hooked on the saddles. As the engine is lowered again and its weight taken by the beam, the height of the spring balance supports should be adjusted to maintain the beam in a horizontal attitude. Then, by taking moments about the supports, the position of the centre of gravity relative to the front and rear ends of the engine may be calculated from the readings of the spring balance less the initial readings due to the weight of the slings and saddles. Another method of determining the centre of gravity position is to employ a three point suspension. Then, from the load at each point, the position of the c.g. in two dimensions may be calculated.

The first method may be used to determine both the height of the centre of gravity and its lateral position relative to appropriate datum planes. To obtain these positions, the engine must, of course, be suspended with its longitudinal axis in the same vertical plane as before but turned through 90 deg so that the cylinder axes are horizontal, and again with its longitudinal axis horizontal and at right angles to the axis of the beam. The apparatus required for centre of gravity determination should be part of the permanent equipment of the development department.

Fig. 3. Details of the Dodge rear mounting

Having determined one point through which the axis of oscillation of the engine passes, the next step is to locate the axis positively by determining its angle of inclination to the horizontal. This axis would be a principal axis of inertia of the engine and gearbox unit if the torque impulses acted in a direction parallel to it. In fact, the two axes are not parallel, but errors resulting from the assumption that they are will be small, and may be ignored.

Nevertheless, a background of knowledge as to what actually happens is always useful, since it enables the designer to foresee and understand phenomena, the occurrence of which might otherwise take him by surprise. A mass has three principal axes of inertia which are mutually perpendicular. In engine mounting design we are concerned mainly with the two axes in the longitudinal vertical plane. The mass moment of inertia about these axes is in one case a maximum and in the other a minimum, and a polar diagram of the moments of inertia about all other axes passing through the same origin, that is the centre of gravity, is an ellipse. In other words, the two principal axes of inertia are the major and minor axes of the ellipse.

It follows that if the mass is rotated about an axis other than the minor principal one, centrifugal force will impose on it a couple tending to turn the mass until it does rotate about a principal axis, Fig. 1. Moreover, to accelerate the mass about any axis other than the major principal axis through the centre of gravity, a greater amount of energy will be required. The least moment of inertia is that about the major axis of the ellipse, and when rotating freely under the influence of a couple, the mass will take the course of least resistance. If a mass is accelerated about any axis other than one through the centre of gravity, the inertia of the portion of the mass on one side of the axis will be larger than that on the other side. As a result the part on the lighter side will accelerate and move faster than that on the heavy side; this amounts to a tendency to rotate about the centre of gravity.

If the direction about which the torque acts is not parallel to a principal axis, other influences which modify the motion are brought to bear. The applied torque may be resolved into two components, one about the nearest principal axis and the other about an axis at right angles to it, that is, about the other principal axis. If, as is often the case with a conventional engine and gearbox arrangement, the torque axis and the principal axis are separated by only a relatively small angle, the component about the principal axis adjacent to the torque axis will be by far the greater one. Thus the effect of the other component will be so small that it may be ignored without introducing appreciable errors as a result.

Another reason why the effect of the angular offset of the torque axis is negligible is that the magnitude of the torque fluctuation is small relative to the mass moment of inertia. It is easy to understand that a large torque acting on a body having a small moment of inertia will have a much more overpowering effect than if the order of magnitudes is reversed. It follows that, because the magnitude of the out-of-

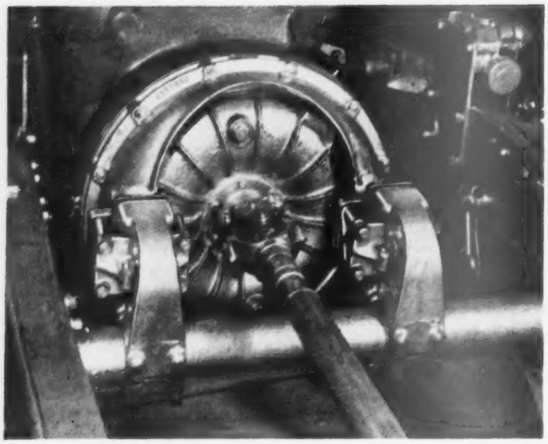

Fig. 4. Lateral adjustment in the rear mounting assembly common to the Daimler CD 650 and CVG 6 is provided for accommodating the different engines

balance forces increases with speed, the deviation of the axis of oscillation from the principal axis also varies with engine r.p.m. Again, this deviation is small and may be ignored.

Before the method of determining the angular position of the principal axes is described, a belief that is fairly widely held must be discredited. This belief, which is erroneous, is that the axis about which the engine oscillates is one passing not only through the centre of gravity, but also through the centre of the universal joint at the rear of the gearbox. It arose from the fact that many early patent specifications concerning the flexible mounting of engines incorporated this arrangement. Experience has shown that the angle through which the propeller shaft oscillates, as a result of the joints being offset from the axis of oscillation of the engine, is too small to cause torsional fluctuations of any appreciable magnitude. The method of determining the position of the principal axes of inertia in the longitudinal vertical plane is first to find by experiment the mass moments of inertia about any two axes mutually perpendicular in the same plane. Then the moment of inertia about an axis bisecting the angle between the other two must be determined.

Let J_Y and J_X = the moments of inertia about the two principal axes, YY and XX

θ = the angle measured from any other pair of mutually perpendicular axes, xx and yy to XX and YY respectively

J_x and J_y = the moments of inertia about the axes xx and yy

J_z = the moment of inertia about an axis zz at an angle of 45 deg to xx and yy.

Then, by substituting in the following expression the values obtained experimentally for J_x, J_y and J_z, the angle θ may be found.

$$\text{Tan } 2\theta = \frac{2J_z-(J_x+J_y)}{J_x-J_y}$$

The values J_X and J_Y may be obtained by substituting the known values in the following pair of simultaneous equations and then solving them.

$$J_X+J_Y=J_x+J_y$$
$$J_X-J_Y=(J_x-J_y)\text{ Sec}2\theta$$

In order to avoid further complication, the mathematical derivation of these formulæ has been relegated to the appendixes.

Two methods of determining the mass moments of inertia about the arbitrary axes xx and yy, and zz will be described. Both must be carried oût as accurately as possible, since the final result obtained by the calculations given above is the difference of two moments of inertia. This difference may in some cases be very small, and an error of small magnitude may represent a large percentage of the final answer.

The first method is based on the compound pendulum theory. Knife edge supports are used to carry the power unit in such a way that it may swing about any axis, above and parallel to the arbitrary axis xx, which passes through the centre of gravity. The engine, thus supported, is set in motion so that it swings backwards and forwards, care being taken that the angle of swing is not too great. This precaution is necessary since the theory is based on the assumption that the angle of swing is small. The time taken to complete a number of oscillations is checked with a stop watch, and hence the time taken to complete one is calculated. The larger the number of oscillations timed, the greater is the accuracy of the final result.

Let T = the time in seconds for one complete oscillation

k = the radius of gyration about axis xx

d = the distance between the axis of oscillation and the axis xx

$g = 32{\cdot}2$ ft/sec²

Then, $T = 2\pi\sqrt{\dfrac{k^2+d^2}{gd}}$

or, $k^2 = 0{\cdot}815T^2d - d^2$

Then, if J_x = the moment of inertia about the axis xx

and W = the weight of the power unit,

$$J_x = \frac{W}{g}k^2$$

This procedure is repeated to find the moment of inertia about the axis yy and again about zz. It is probably most convenient to choose the axes xx and yy such that they are parallel and perpendicular to the axis of the crankshaft. Then a wide choice of reference planes is available when setting up the engine for the experiments.

The second method that may be employed to determine J_x, J_y and J_z is to suspend the power unit, on either two or three light wires, first in such a manner that the axis xx is vertical. The unit is caused to oscillate about this axis, and the time taken to complete a large number of oscillations measured. Then this is repeated for the axes yy and zz, and the times for one oscillation are calculated in each case.

Let W = the weight of the unit, in lb

L = the length of the suspension wires, in ft

R = the radial distance between each suspension wire and the axis of oscillation, in ft

α = the angle between the wires and the vertical

Fig. 5. The Daimler underfloor engine mounting arrangement

θ = the angle of twist of the unit from its static position, in radians.

Then, the restoring force for a small deflection is:

$$F = W \sin \gamma = \frac{WR\theta}{L}$$

The restoring couple $M = \frac{WR^2\theta}{L}$

and the restoring couple per unit displacement

$$C = \frac{M}{\theta} = \frac{WR^2}{L}$$

$$\text{Now, } T = 2\pi\sqrt{\frac{J}{C}}$$

$$\text{or } T = 2\pi\sqrt{\frac{JL}{WR^2}}$$

$$\text{Hence } J = \frac{WR^2T^2}{39{\cdot}5\,L}$$

Again care must be taken to ensure that the oscillations are of small amplitude.

All the basic information necessary to determine the layout of the system is now in hand. The general principles to be followed are that the mountings must be so positioned that the unit is free to oscillate about its appropriate principal axis; it must also be free to vibrate in the direction of the reciprocating out-of-balance forces; and it must be positively located against fore and aft motion which might interfere with the operation of clutch and other controls.

Under resonant conditions the engine and gearbox assembly will tend to oscillate, as has already been explained, about the principal axis. If it were not mounted in such a manner that it could rotate freely about that axis, lateral forces would be applied to the mountings. Thus, one of the principal reasons for adopting this arrangement is to avoid these undesirable lateral vibratory forces.

The actual arrangement adopted will almost certainly be determined by the layout of the vehicle structure. Nevertheless, the fact that engine mountings must be catered for should always be borne in mind when designing the structure. In that way, problems may often be circumvented at an early stage in the vehicle design, whereas later they might not be solved so easily or so satisfactorily.

In general, the greater the radius from the principal axis to the mountings, the softer must be the rubber, and the more difficult it is to obtain satisfactory isolation against torsional vibrations. The same principle applies so far as longitudinal spacing of the mountings is concerned. The longitudinal spacing determines not only the degree of isolation obtainable from rocking couples about the transverse plane of symmetry of the cylinders, but also from that obtainable against couples due to the offset of the axis of percussion from the centre of gravity of the power unit. The axis of percussion is a term often used to denote the line of action of the reciprocating forces.

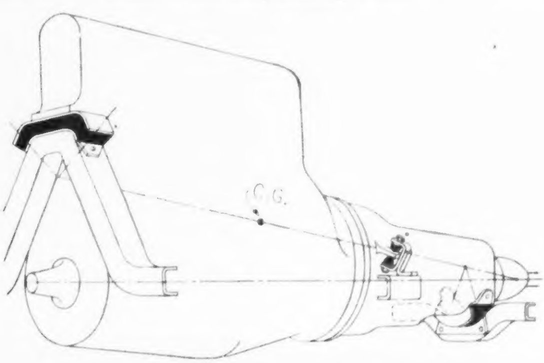

Fig. 8. The Chrysler Floating Power System

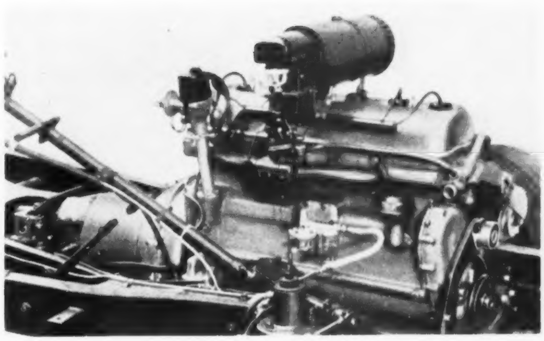

Fig. 6. A high front mounting point is employed on the Armstrong Siddeley Sapphire

Opinions vary on the subject of the correct longitudinal disposition of mountings. However, experience does suggest that the front and rear mountings should take approximately equal loads. In this way neither is required to work under excessively severe conditions, and the requirement with regard to the ratio of vertical to torsional stiffness is reasonable.

In some current and past designs, in which the rear mounting is a long way aft of the centre of gravity, it would hardly seem possible that adequate flexibility can have been incorporated to ensure effective isolation of the structure from reciprocating out-of-balance forces and couples. However, this arrangement may have been, in some cases, difficult to avoid, since the layout of the vehicle structure has made it difficult to carry the rear mounting at a point further forward.

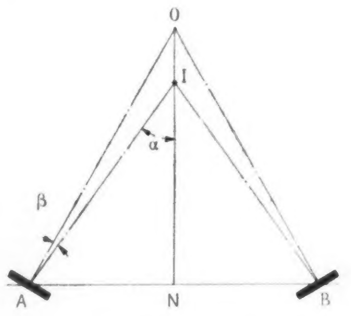

Fig. 7. Arrangement of Vee mountings

Certain types of mountings should not be set in a vertical transverse plane, and rarely, if ever, should any type be positioned normal to the axis of the crankshaft. A little thought on this subject will indicate that the plane in which the mountings are set ought to be, if possible, normal to the axis of torsional oscillation; and the stiffness in a direction parallel to the axis of the cylinders must be such as to give adequate isolation of the structure from reciprocating forces.

However, there are certain circumstances under which the mountings may be set vertically. When sandwich type front mountings are employed they are often arranged in a transverse vertical plane in order that they may be more effective in supporting the weight of the engine. If the sandwiches were set at an angle to the longitudinal horizontal axis there would be a tendency, which would have to be reacted at the rear, for the engine to slide backwards down the inclined plane of the mountings. This layout in the transverse vertical plane is justified in the following manner.

The oscillations of the engine about the principal axis may be resolved into two components, one about a horizontal axis and the other about a vertical one. With the mountings positioned in the vertical plane, the component about the horizontal axis is properly catered for. If a horizontal sandwich is employed directly below the axis at the front, the component about the vertical axis is also adequately isolated from the vehicle structure. However, with a Vee arrangement of sandwich mountings, this component is reacted partly by shear in the rubber and partly in compression. If the proportion reacted by compression is small, the arrangement in the vertical plane is evidently satisfactory, since the majority of manufacturers have adopted it. The method of calculating the correct Vee angle is described at the end of the section on commercial vehicles.

The setting in a transverse plane at an angle from the vertical in certain cases may have another advantage in that

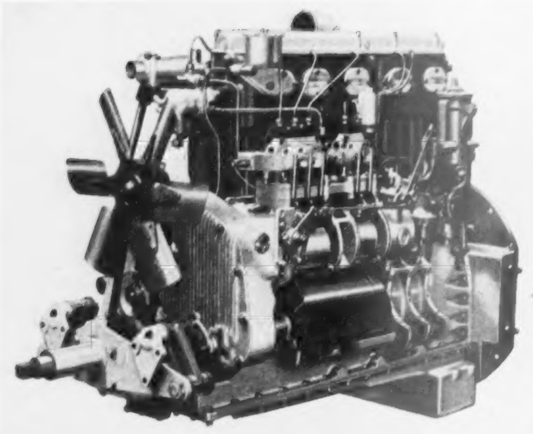

Fig. 9. A rubber bushed link type of front suspension is used on the Gardner engine installation in the E.R.F. chassis

it leads to a greater degree of firmness in reacting inertia loads due to braking or acceleration. Apart from the angular setting, positive longitudinal location may be built into the mountings. This will be discussed later in connection with individual mounting unit details.

Torque stops are sometimes incorporated to limit the amplitude of oscillatory motion of the engine under extreme conditions of resonance, and to limit its travel at full torque or when the clutch is engaged violently. Such stops, to some extent, defeat the object of fitting flexible mountings. They are rarely necessary with six cylinder engined cars and even with four cylinder units; it is better where practicable to allow the engine to move freely over its whole range of travel. However, stops are nearly always required on trucks, because of the low gear ratios employed.

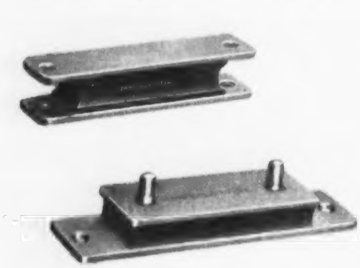

Fig. 10. Examples of plain rubber-to-metal bonded sandwich type mountings

Commercial vehicles

Commercial vehicle mountings have to satisfy the requirements already mentioned, and some others as well. Fleet operators are much more cost-conscious so far as maintenance is concerned than private car owners. As a result engine removal must be easily effected. Moreover, it should be possible to remove the gearbox without taking out the engine. Another characteristic of commercial vehicles that affects the engine mounting is the large frame deflections.

To meet all these requirements a single, low front point is often used in conjunction with two rear points, one on each side of the engine crankcase, Figs. 2 and 3. This arrangement has the additional advantage that it is generally easily adapted to conventional frame designs. A disadvantage is that the rear mountings may be somewhat heavily loaded. However, sometimes the two rear mountings are positioned one on each side of the gearbox. When different types of gearboxes or alternative engines are available to suit customers' requirements, a means of adjustment is often provided to alter the lateral position of the rear mountings. The installation of the Daimler CD 650 and C.V.G.6 power units is an example of adjustable mountings at the rear to accommodate different engines, Fig. 4.

Changes to a different type of gearbox, and indeed any other major change, may alter the position of the axis of oscillation. Therefore in all designs, commercial or private car, types of mounting most adaptable to changes of axis of oscillation should be employed if possible. Sandwich mountings are probably best and rubber bushed trunnions worst in this respect.

In general, the vibrational impulses developed by commercial vehicle engines are, of course, larger than in passenger cars. However, the mass and inertia of the engines is greater, and thus the mountings are made of a more robust design, so they are automatically capable of withstanding the larger vibratory loads.

In diesel engines, unlike petrol units, the magnitude of the impulses does not decrease as engine torque is reduced. This is because of the method of governing these engines by controlling the rate of fuel supply. Troublesome frequencies lower than the second order are rarely experienced, since the distribution and therefore firing is much more even than in petrol engines. The range of frequencies to be catered for is smaller in diesel units because of their lower maximum speeds. Torque fluctuations are large at idling and when pulling away at low speeds.

So far as flat underfloor engines are concerned, the fundamental requirements are the same as for vertical engines. The fact that these requirements ar not met in some designs might account for the high noise level occasionally experienced in this type of vehicle. The main difference between the underfloor and vertical layouts is that with the flat engine the principal axis is horizontal or nearly so, but it often is not parallel to the longitudinal axis of the vehicle.

An example of a current underfloor engine design is the Daimler, Fig. 5. At the front, a Vee arrangement of two double sandwich units is employed. The lower plate of each of the two sandwiches is supported inside the bracket which, as can be seen in the illustration, has on its top face five vertical studs. The two upper plates are carried on separate brackets on the engine. At the rear, two similarly arranged double sandwich units may be seen in the illustration. Between them, to provide axial restraint, is a tie rod rubber bushed at each end. Both the lower plates and the rear eye of the tie rod are carried on a suitably shaped cross piece. The rear mountings are higher than those at the front, but both pairs are focused on a point on the axis of oscillation.

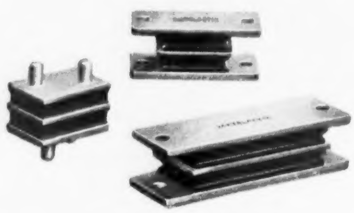

Fig. 11. Some bonded sandwich type mountings with a centre plate for stabilizing the rubber portions of the units

It must be emphasized that in focusing the sandwich type mountings, allowance must be made for the ratio of compression stiffness : shear stiffness. This may be done as follows. In Fig. 7, A and B are the two sandwich mountings viewed from the front, O is the point of intersection of two lines perpendicular to the plane of each sandwich drawn from the centre of the mountings, and I is the instantaneous centre in the transverse plane in which the mountings are set. Another line is drawn through OI and projected until

Fig. 12. A selection of Metalastik vertical sandwich units with a load carrying centre plate

it meets AB at N. Thus IN is perpendicular to AB. Let angle AIN $= \alpha$, and angle IAO $= \beta$. Then the geometry of the system must be such that $\tan \beta \times \tan (\alpha - \beta) = 1/k$, where $k =$ the ratio of compression stiffness: shear stiffness.

Front mountings

Because of the need to support the weight of the engine, and since translational inertia is greater than rotational inertia, all mountings must be appreciably stiffer in the vertical direction than torsionally. If this were not so, unduly large deflections would be obtained. Fortunately, greater stiffness in the vertical direction still results in satisfactory vibration isolation, because the vertical frequencies are of the second order and are appreciably higher than the half order torsional vibrations likely to be experienced. One way of providing these characteristics is to carry the engine on rubber bushed trunnions, one at the front and one at the rear, the axes of which are in line with the axis of oscillation. This arrangement is not often possible at the rear, but is sometimes employed at the front. One of the latest designs incorporating this feature at the front is the Armstrong Siddeley Sapphire, Fig. 6.

This type of mounting was the subject of several patents in the 1930's. Probably the Chrysler Corporation were the first in the field when they introduced the well known Floating Power arrangement, Fig. 8. The main disadvantage of this system is that it calls for a high mounting point at the front which, because it is mid-way between the two sides of the engine, calls for long and often slender arms to support it. Moreover, it is not always possible to incorporate the mounting in that position, because of the close proximity of the fan, water pump, and drive. With this type of mounting it is, of course, essential to provide some form of positive longitudinal location. This is usually done at the rear mounting.

Fig. 14. Some slotted and plain Metacone units

In this country, Metalastik Ltd. have for many years manufactured under licence Floating Power engine suspension units, and further developed the idea. The first rubber bushed trunnion mountings, although they had adequate flexibility to absorb torsional vibrations, were far from ideal as a protection against the undesirable effects of vertical or transverse oscillations. If a quantity of rubber large enough to give characteristics satisfactory in all respects was incorporated, the mounting was generally too large to be accommodated in the confined space available.

The first logical development was to use a half of a larger bush that had been in effect split diametrically. This semicircular segment was positioned below the trunnion pin,

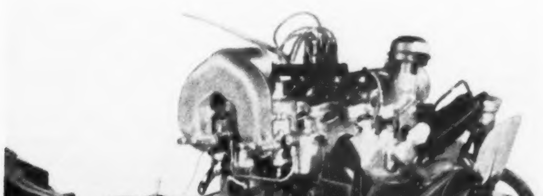

Fig. 15. In the Rover 75 two Metacone mountings are fitted at the rear, with their axes in a transverse plain perpendicular to the axis of oscillation

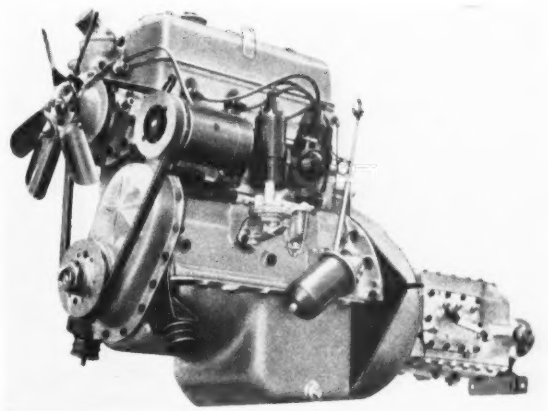

Fig. 13. The rear of the Lanchester 14 engine and gearbox unit is carried on the centre plate of a vertical sandwich mounting

and in this way the top half, which did very little work in any case was eliminated. The next step was to split the half bush vertically to make two quarters which could be positioned at any convenient radius below and to each side of the axis of oscillation.

Yet another simple modification of this system was the use of a plain horizontal rubber sandwich directly below the axis of oscillation. Later it was realized that such a mounting could be used much lower down where it could be secured conveniently to the crankcase or sump. In this position the weight of the engine, and vertical out-of-balance forces are taken by compression. On the other hand, torsional vibrations, which give rise to transverse motions of parts of the engine immediately below the axis of oscillation, are taken by the rubber shear. The flexibility in shear and compression of this type of unit may be increased by using a thicker rubber insert, with one or more metal plates included in the sandwich to prevent excessive bulging of the rubber under compression.

The principal disadvantage of all these sandwich type mountings, where the weight of the engine is taken by the rubber in compression, is that rebound loads are taken in tension, and when bonded components are used this is not a particularly desirable feature. It is a factor to be carefully considered when designing vehicles to be used on rough terrain, either in this country or overseas, but is not so important for passenger cars designed for reasonably good roads. Rebound loads may be catered for by incorporating rebound stops in the mountings. However, this inevitably makes it more expensive, and as a general rule the simpler mountings are more effective.

An alternative, giving equally good support for both bump and rebound loads, is to suspend the engine from two links, one on each side. Each link has a rubber bushed eye at both ends. The upper end is trunnion mounted on the vehicle structure while the lower end carries the engine. Under static conditions the links may be more or less horizontal, so that the weight of the engine is supported mainly by the bushes in torsion. However, bump or rebound loads cause additional deflection and are taken mainly by tension in the links and compression in the bushes. Thus the unit has a variable rate which increases with the load.

This arrangement again is more expensive than most. However under circumstances such as might arise in chassisless vehicles where the main structure is high relative to the engine, the link type unit could be less expensive than incorporating a cradle-like member to carry more

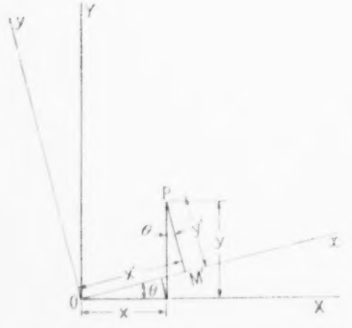

Fig. 16. Diagram for Appendix 1

conventional mountings. It has an advantage over the single high point trunnion mounting in that a greater thickness of rubber is available to take compression loads, and the links are more easily accommodated on each side of the engine.

A twin link front mounting is shown in Fig. 9. In this example, which is the Gardner engine installation in the E.R.F. chassis, the links are more nearly vertical, and will therefore give a more positive reaction to heavy bump and rebound loads. At the rear, a single horizontal bush is employed with its axis directly above the axis of oscillation.

Rear mountings

The principles on which rear mounting layouts are based are much the same as those applicable at the front. However, in many cases, the principal exception being in some commercial vehicles, the static load is not so great. Moreover, fore and aft location is often effected by the rear mounting.

It is rarely, if ever, possible to employ at the rear a single rubber bushed trunnion with its axis in line with the axis of oscillation. Nevertheless, one or more of these units are often incorporated with their axes in a transverse plane and at right angles to a radial line between the axis of oscillation and the mounting point. If one unit alone is fitted, it must be either directly above or below the axis of oscillation.

Thus, torsional loads are reacted by axial shear in the bush, while vertical and longitudinal loads are taken in compression. It is possible by removing some of the rubber from appropriate parts of the bush to make it stiffer in compression in one direction than in another. The greatest stiffness is required in the direction of the longitudinal axis of the engine. When two bush type units are fitted, the Vee layout is usually adopted. With this arrangement torsional loads are reacted by axial shear while vertical ones are reacted partly by shear and partly by compression. By the use of a suitable Vee angle, the correct ratio between the stiffness in each direction is obtained.

A horizontal sandwich type mounting Figs. 10 and 11, is sometimes employed at the rear but, in addition to the troubles that might arise from rebound loads, it has the disadvantage that further measures must be taken to provide fore and aft location. The same objection with respect to fore and aft location applies to a simple sandwich unit mounted vertically in a transverse plane.

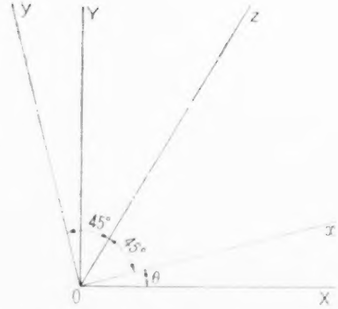

Fig. 17. Diagram for Appendix 2

A more satisfactory arrangement is the vertical double sandwich unit with the engine supported by the centre-plate, Fig. 12. Both torsional and vertical loads are taken by the rubber in shear, while the fore and aft loads are taken in compression. It should be noted, however, that when the unit is in a plane normal to the axis of oscillation, it is not parallel to the axis of percussion. Thus, there will be a small component of the reciprocating out-of-balance forces taken by compression in the rubber. A compressive pre-load may be applied to counter the tensile stresses complementary to the shear in the rubber. This has the effect of increasing the life of the unit. The Lanchester Leda, Fig. 13, is an example of a current design incorporating this feature.

Cone type mountings are often employed at the rear, and occasionally at the front. With these units, Fig. 14, axial loads are taken partly in shear and partly in compression. The proportion taken by compression increases with the load so that the unit has a variable rate, which to some extent obviates resonance problems. These, like the simple bush type units, may incorporate slots in the rubber to give different stiffness rates in different directions. They are usually set, one on each side of the engine, in the plane normal to the axis of oscillation, but with their axes vertical when viewed from the front. This arrangement is employed on the Rover 75, Fig. 15.

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APPENDIX

1. *To find the polar moment of inertia J about any axis, given the principal moments of inertia, Fig 16.*

Let J_X and J_Y be the principal moments of inertia. Let J_x and J_y be the moments of inertia about any other axis Ox and Oy.

Then $y' = PM$

$= y \cos\theta - x \sin\theta$

and $x' = x \cos\theta + y \sin\theta$

If δA is the element at P, then

$$J_x = \int y'^2 dA$$
$$= \int (y\cos\theta - x\sin\theta)^2\, dA$$
$$= \cos^2\theta \int y^2 dA - 2\cos\theta\sin\theta \int yx\,dS + \sin^2\theta \int x^2 dA$$

but $xy\delta A = 0$, since OX and OY are the principal axes,

therefore $J_x = J_X \cos^2\theta + J_Y \sin^2\theta$ (i)

Similarly, $J_y = J_X \sin^2\theta + J_Y \cos^2\theta$ (ii)

add (i) and (ii)

$J_x + J_y = J_X + J_Y$

2. *To find the principal axes, Fig 17.*

When a mass is without an axis of symmetry, the position of the principal axes may be determined as follows:
Find the position of the centre of gravity and take any pair of mutually perpendicular axes Ox and Oy through it. Let OZ bisect the angle yOx, and let OX and OY be the required principal axes. Then J_x, J_y and J_z equal the polar moment of inertia about OX, OY and OZ, and J_X and J_Y equal the polar moment of inertia about OX and OY.

$$J_z = J_X \cos^2(45+\theta) + J_Y \sin^2(45+\theta)$$
$$= \tfrac{1}{2} J_X [1+\cos(90+2\theta)] + \tfrac{1}{2} J_Y [1-\cos(90+2\theta)]$$
$$= \tfrac{1}{2} J_X [1-\sin 2\theta] + \tfrac{1}{2} J_Y [1+\sin 2\theta]$$
$$= \tfrac{1}{2}(J_X+J_Y) + \tfrac{1}{2}(J_Y-J_X)\sin 2\theta$$
$$= \tfrac{1}{2}(J_x+J_y) + \tfrac{1}{2}(J_Y-J_X)\sin 2\theta$$

$2J_z = (J_x+J_y) + (J_Y-J_X)\sin 2\theta$ (iii)

Also, by subtracting (i) from (ii):

$$J_y - J_x = J_X \sin^2\theta + J_Y \cos^2\theta - J_X \cos^2\theta - J_Y \sin^2\theta$$
$$= J_X(\sin^2\theta - \cos^2\theta) + J_Y(\cos^2\theta - \sin^2\theta)$$
$$= (J_Y - J_X)(\cos^2\theta - \sin^2\theta)$$

$J_y - J_x = (J_Y - J_X)\cos 2\theta$ (iv)

but from (iii):

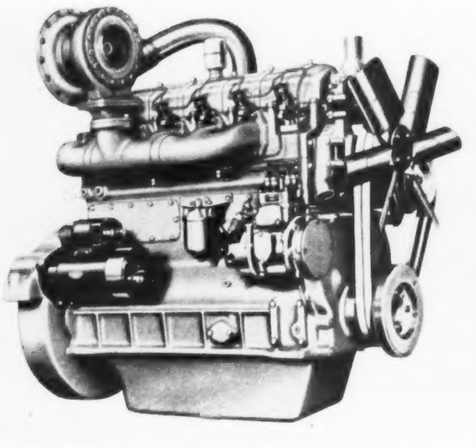
$(J_Y - J_X)\sin 2\theta = 2J_Z - (J_x + J_y)$ (v)

Divide (v) by (iv)

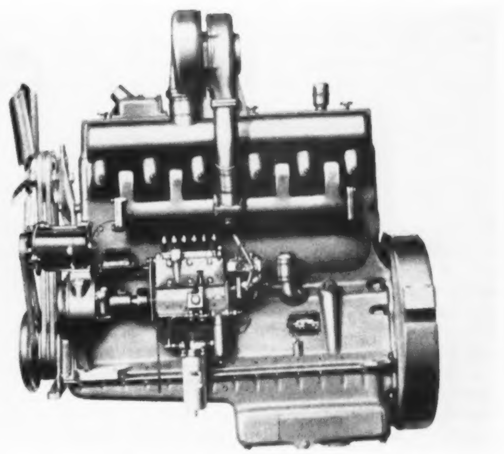
$$\operatorname{Tan} 2\theta = \frac{2J_Z - (J_x+J_y)}{J_y - J_x}$$

TURBO-CHARGED DIESELS

New Engines Developed by Waukesha Motor Company



185 h.p. Waukesha engine with Schwitzer-Cummins turbo-charger



352 h.p. Waukesha diesel with Elliot turbo-charger

FIVE engine sizes are now being produced by the Waukesha Motor Company, Waukesha, Wisconsin, U.S.A., as turbo-charged units and the range includes relatively small engines. This Company was one of the first to exploit the fairly recent availability of commercially produced turbo-chargers of small size. They also claim to be first engine builders to produce a turbo-charged engine with only 426 in^3 displacement.

Turbo-chargers have been applied to five basic engine sizes, three suitable for transport or industrial service and the two largest as complete units for industrial service. All the engines are heavy-duty, six-cylinder, four-stroke cycle units. The three transport models have displacements of 426, 779 and 1197 in^3. Respective stripped engine maximum powers are: 185, 280 and 352 h.p. The equivalent maximum powers without turbo-charging are: 147, 200 and 225 h.p.

All the engines incorporate the Waukesha spherical fuel combustion chamber. The design of the spherical cavity and the tangential throat opening into the main combustion chamber promote controlled turbulence and help in obtaining clean, complete combustion. An insulating air space surrounds the lower half of the combustion chamber, and the retained heat is given up to the air during compression. This reduces ignition lag, and since the amount of heat retained varies with the load, this feature has the effect of advancing or retarding the time of combustion. The upper half of the chamber, including the injector mounting bore, is water jacketed.

American Bosch injection equipment suited to the engine size is fitted to all Waukesha turbo-charged diesels. Single orifice, pintle-type nozzles are used. The smallest engine has a single-plunger, flange - mounted injection pump. The others have multi-plunger, bracket - mounted pumps. Variable speed centrifugal governors, mounted on the injection pumps, are used on the three transport models. The larger engines have centrifugal type governors operated through the engine gear train.

Two designs of turbo-charger are used. The smallest engine has the Schwitzer-Cummins turbo, while the Elliot turbo is standard equipment for the other four models. On the three smaller engines the turbo is ordinarily top-mounted on the exhaust manifold. On the two larger units the turbo is mounted on the rear end of the exhaust manifolds. At top engine speeds, the rotating elements of the turbo-chargers turn at a speed that may be as high as 40,000 r.p.m.

All the special features that contribute to the long life and reliability of Waukesha diesels are incorporated in the turbo-charged models. The cylinder liners are of special alloy; the connecting rods are heat-treated drop forgings; and the pistons are of heavy duty design with chromium plated wedge-type top rings. Precision type, replaceable bearings are used. The water and oil pumps are built-in types, and Stellite faced valves and Stellite inserts are standard. Depending upon the service requirements the drop forged, heavy crankshafts, with large diameter hardened and ground journals, may be supplied with or without counter-balancing. Full pressure lubrication is common to all models.

The advantages of using a turbo-charger to utilize the waste energy in exhaust gases for supplying a denser charge of air to the engine have long been recognized. However, this has not been possible hitherto, with transport engines, because until fairly recently small turbo-chargers were not produced on a commercial scale. Furthermore, older designs of turbo-chargers were not particularly responsive to speed changes, and as a result the acceleration characteristics of the engine suffered. Current design improvements, involving lighter rotating parts, improved bearings, better metals and more efficient turbines and compressors have largely solved this difficulty. Primarily, turbo-charging is adopted to obtain a higher specific output from the engine. This means a better horse-power/weight ratio, increased mechanical efficiency and therefore improved fuel efficiency.

Lightweight Bogie

IN the February 1953 issue of this journal, the article entitled "Tippers and Trailers" included a description of a non-detachable bogie with a special suspension system, so designed that all the braking strains are taken by the ends of the leaf springs which are enclosed in a box that contains a rubber mounting. Unfortunately, this design was ascribed to the wrong manufacturers. Actually, the bogie in question was designed and manufactured by R. A. Dyson and Co., Ltd., 76-78, Grafton Street, Liverpool. We now understand that the rubber mounted springs are exclusive to Dyson production.

THE GAS TURBINE CAR

A Review of Problems Still to be Solved

By F. R. Bell

CONSIDERATION of the problems involved in the development of a successful gas turbine car shows many possible difficulties. However, except for the problem of fuel consumption, most of the difficulties that have been suggested are imaginary. This article sets out to show in what way both the turbine and the car as we know them must be modified and how they must be matched together. Unfortunately in order to do this, and to make the reasons clear, it will be necessary to go briefly into certain aspects of both car and turbine design and theory.

Conveniently we can discuss this under three main headings, first the problems associated with the turbine, second the transmission and additions necessary to make the modified turbine suitable for the car and last the way in which the car as we know it must be re-designed to suit the needs of the turbine.

Most of the information contained herein is the result of work done by the Rover Company, to whose initiative and engineering ability, the eventual success of the gas turbine car will, in no small measure, be due. A few words can also be said, although they are really outside the scope of this article, on the question of how the Motor Industry might have to alter its production and servicing set-up when the gas turbine car arrives.

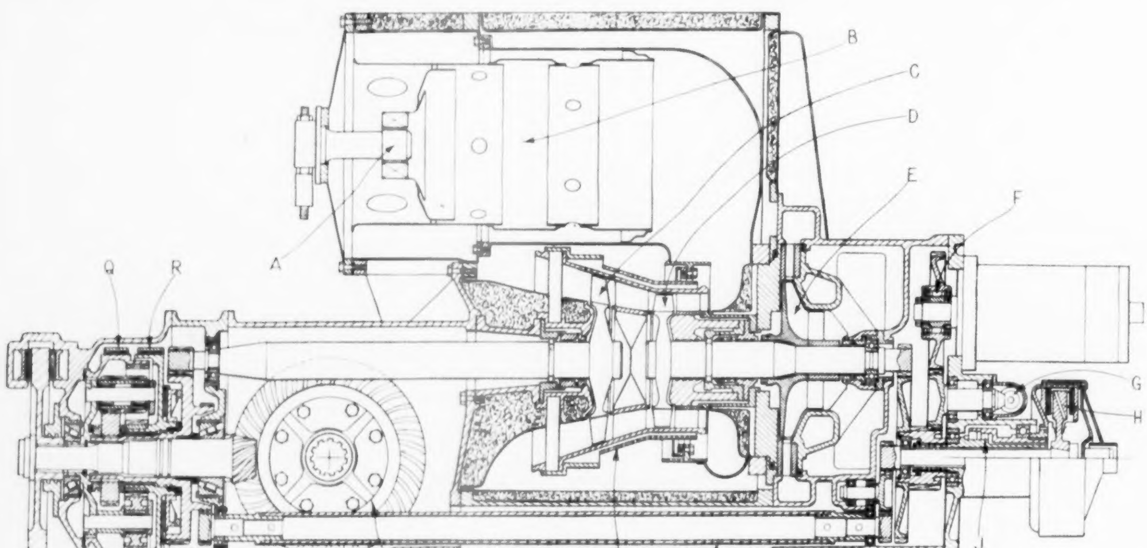
The gas turbine

Only the open cycle true gas turbine using air as a working fluid, will be considered here. A discussion of the relative merits of alternative arrangements such as mixed cycles i.e. steam (or other fluids) combined with the gas turbine cycle, closed cycles using air or other fluids and so-called gas turbines using positive displacement compressors or turbines, or pulse systems would be so involved as to require a separate article.

First and most important is the specific fuel consumption of the turbine since it is in this respect that the present small turbines are much inferior to the piston engine. Three main lines of attack are open. By far the best method is to obtain sufficiently high component efficiencies so that the overall efficiency is high, but this is impossible in practice. If the efficiencies obtained by the best large aero turbines could be obtained on a small car turbine, and it should be possible to come fairly near this, then a car without heat exchangers and using only a single stage compressor and single stage compressor and power turbines could give perhaps 12 miles per gallon at all speeds. It should be mentioned here that in the whole of this article a car turbine will be assumed to have a separate power turbine to drive the car as in the Rover unit, since one of the most important advantages of the gas turbine for the car is the torque characteristic of this type. Without a separate power turbine the torque characteristic would be so bad that a very complex gear box would be required, much worse than in the normal car.

Although it has been discussed before by several writers, it is as well to reconsider the effects of component efficiency on the overall efficiency. Perhaps the easiest way to show this is to take an actual case. The following table shows how a 5 per cent increase in turbine efficiency gives a very much larger increase in overall efficiency and that this increase is most marked at the lower powers, that is, at the powers used during a large part of the car's service.

Compressor r.p.m.	27,000	50,000
Total Turbine b.h.p.	25·35	275·8
Required Compressor b.h.p. ..	20·6	188·0
Output Available b.h.p.	4·75	87·8



A Burner
B Combustion chamber
C Power turbine
D Compressor turbine
E Compressor impeller
F Starter free wheel
G Tachometer drive
H Friction clutch
J Free wheel
K Regenerative braking shaft
L Shroud ring cooling air
M Differential
N High-ratio clutch
P Disc brake
Q Reverse clutch
R Low-speed clutch

Fig. 1. Elevation of 100 b.h.p. turbine unit with transmission

A 5 per cent increase in turbine efficiency and thus in turbine power available results in the following:

Increase in Total b.h.p.	1·27	13·79
Output Available b.h.p.	6·02	101·59
Per cent Increase b.h.p.	26·7	15·7

This gives an overall increase as compared with the turbine efficiency increase of over 5 times at low powers and over 3 times at high powers. As the compressor efficiencies cause a somewhat similar effect, it can easily be seen why. It is because in the small turbine these are both down by between 5 and 10 per cent as compared with the best large aircraft turbines, that the overall efficiency is low. It has been suggested by some authorities that these are fundamental scale effects and cannot be avoided but tests have shown that while there certainly is a scale effect it is actually not very large and there is a good possibility of a big improvement in the present position.

As has been mentioned, the best that can be expected from improved component efficiency is not sufficient; this brings us to the most important contribution to fuel consumption, the heat exchanger. Because of the low pressure ratios used in the gas turbine, the expansion ratio in the turbine and hence the temperature drop, which is a function of expansion ratio is low. This results in a high exhaust temperature with a consequent wasteful rejection of considerable quantities of potentially useful heat. A heat exchanger placed in this exhaust gas stream can be used to transfer some of this heat into the air before combustion, so that less fuel is required to heat the air to the maximum combustion temperature usable by the turbine and a saving in fuel results.

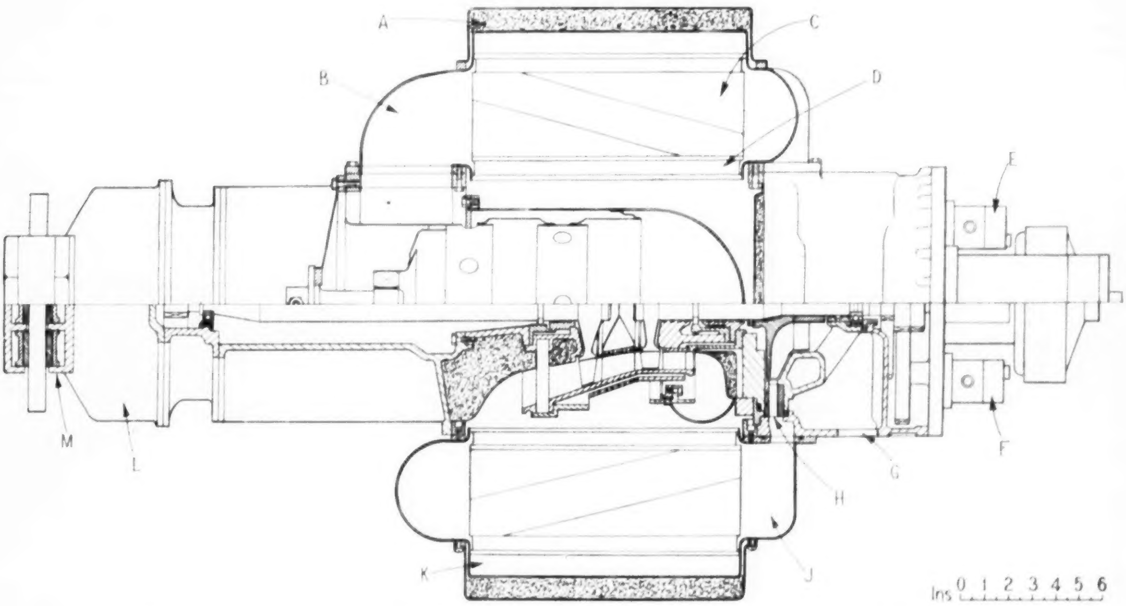
The main difficulties with this system are twofold. First, it is difficult to transfer heat without having either a large surface or a high pressure drop, which cannot be tolerated owing to the loss of power it causes. It is therefore necessary to pack large areas of thin sheet metal or tubes into a small space without creating any restriction or awkward corners to cause loss. This, in practice is an extremely difficult problem and has not yet been really satisfactorily solved, although the solution seems reasonably near. Second, if a unit that has reasonably high heat transfer and low losses is developed, it may suffer from thermal stresses which cause cracking. This is easily cured by making the matrix thick enough but the unit is then heavy and expensive. Attention to detail, however, has resulted in a big improvement in life and there is every hope that continued development will produce adequate life.

Owing to the decrease in expansion ratio with a decrease in speed, the temperature drop in the turbine falls much more than the maximum temperature, therefore the exhaust temperature and hence the temperature rise of the air in the heat exchanger falls less than the turbine inlet temperature. This results in the heat exchanger supplying a greater percentage of the heat required to drive the turbine at low than at high powers. A good heat exchanger might halve the fuel consumption of the engines at full power and reduce it to one-third at low powers. Since it is at low powers that the gas turbine shows the worst comparison with the piston engine, it can easily be seen that the heat exchanger is the real answer to high fuel consumption and primarily it is the development of a satisfactory heat exchanger that is holding up the use of the gas turbine in the automobile at present.

There are three distinct ways of tackling the next problem. This is the time taken to accelerate the compressor unit when the accelerator is suddenly depressed, because it is not until the compressor unit is up to speed that the car accelerates really well. Two of these methods will be dealt with in the next section but as far as the turbine itself is concerned the problem is as follows.

The acceleration time depends on two factors: the excess power available when the maximum fuel flow is fed into the combustion chamber thus producing the maximum temperature; and the resistance to acceleration of the compressor unit which is a function of the moment of inertia of the compressor turbine. The discussion of these is rather involved but a brief outline can be given here.

The more fuel fed into the combustion chamber the higher the temperature and, since the turbine power depends on inlet temperature, the more rapid the acceleration. Maximum fuel flow is limited firstly by the maximum permissible temperature the turbine nozzle blades can withstand and secondly by the fact that an increase in temperature (because the density decreases lineally with temperature while the velocity increase is proportional to the square root of the temperature) causes a decrease in flow which forces the compressor finally on to an unstable part of its characteristic and causes the phenomenon known as surging. Owing to the very short time



A Sound absorber
B Exchanger air outlet
C Heat exchanger
D Exchanger gas inlet
E Oil pump
F Fuel pump
G Air intake
H Compressor delivery
J Exchanger air inlet
K Exhaust
L Gearbox
M Brake cylinders

Fig. 2. Plan of 100 b.h.p. turbine unit with transmission

taken for acceleration the blades cannot follow the temperature changes, so that the limitation is usually caused by the compressor reaching its surge point. Matching further from the surge point results in a fall in efficiency during steady running. The net result is that from this point of view there is very little that can be done to improve acceleration.

As regards moments of inertia the following points are worthy of note. For a given engine power, the highest shaft speed for the turbine blade speed required to drive the compressor results in the smallest turbine and thus in the lowest moment of inertia of the turbine. The compressor, because it can be aluminium, is much less important, but a two stage compressor causes a high turbine inertia because for the same job it will run slower than a single stage. For a given compressor, a two stage turbine, owing to the reduction in required blade speed will result in a reduced moment of inertia. We must, however, weight these points against other considerations.

Another design factor to consider is that the high speed bearings and the compressor turbine rotor will always be likely to have the shortest life. They should, therefore, be easily changed in servicing as is the case with the Rover interchangeable compressor turbine unit.

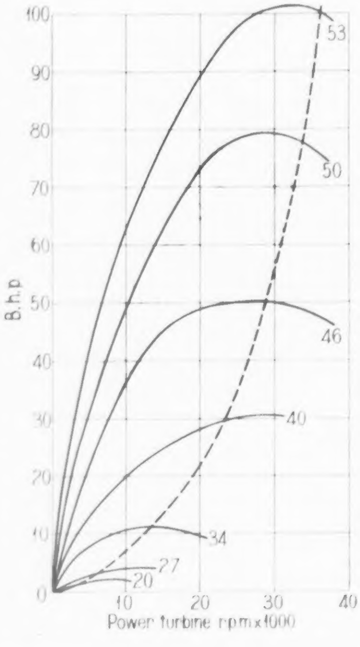
The heat exchangers, also, should be accessible but it may very well be that they will finally have a very long life and will only need blowing out occasionally with compressed air to remove the soot. This is equivalent to decarbonizing a petrol engine. Tests have shown that any soot formation yet encountered is easily removed with compressed air. This completely restores the heat exchanger to its original condition. Experience has also shown that even under bad combustion conditions, and if soot is going to form, a very small soot deposit with an almost negligible effect on performance is rapidly formed; this initial deposit seems to act as a protection and only slow deposition occurs afterwards.

Figs. 1 and 2 show a turbine designed on the lines indicated for about 100 b.h.p. with present turbine efficiencies. Its power curves would be as shown in Fig. 3 and it would be geared so that the car drag curve during steady running matched just beyond peak power causing a drop of a few per cent on efficiency in order to give suitable acceleration conditions. At present an engine of something like this size is required and the air mass flow would be about 2 lb per sec at full power but as efficiencies improve the size could be considerably reduced. The specific consumption of an engine on these lines is shown in Fig. 4 compared with the specific consumption of an engine with component efficiencies which should some day be obtained.

The transmission

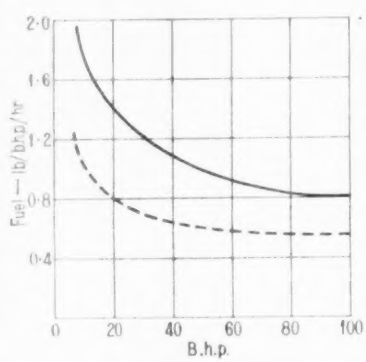
Under this heading are included additions and modifications which are not strictly transmission but which are connected with it, or not strictly part of either the turbine or the car. Although the turbine car can perform quite adequately without any gear changing at all during normal driving, it is still a very debatable point as to whether it should have a two-speed gearbox or not. It certainly should never need more than two for a car and probably not more than three for a bus or lorry.

Although under all normal conditions one speed is satisfactory, if this one speed were used to cover emergency conditions also, the gearing would be such as to limit top speed too severely. It is fairly certain that an emergency low will be needed and it is obvious that a reverse is also needed. If a gearbox containing an emergency low and reverse is designed, a small extra complication can make it such that the low can be automatically changed at a predetermined speed, say 25 m.p.h., thus making an automatic two-speed box that would give the car a somewhat improved performance. Actually the characteristics of a gas turbine are not greatly different from those of a piston engine and torque converter and it has been found by experience in America that for this case a two-speed gearbox is the most suitable arrangement.



Compressor turbine r.p.m. ÷ 1,000
Car drag curve

Fig. 3. Typical power curve for 100 b.h.p. turbine



Best obtainable with present component efficiencies
Attainable with component efficiencies approximately equal to those of aircraft units

Fig. 4. Specific fuel consumption of automobile gas turbine with heat exchanger

The conclusion is that a two-speed and reverse unit almost identical with that of a typical American torque converter car is the best solution. In effect the turbine replaces the piston engine and torque converter. The best gear ratio is around 2, as owing to its flat characteristics a small change is of very little help to a turbine, but 2 is usually difficult with an epicylic box so that either 1·8 or 2·4 should be used. Only experience can show which is preferable but it is likely to be the 2·4 figure.

As it is convenient on manœuvring to swing the lever from reverse to ahead to get the same effect each way with a fixed accelerator position it is useful to keep the same ratio for both ahead and reverse. In order to do this easily and to get easily into reverse or low with the car stationary will require an epicyclic type of box with bands; therefore the general design of the box will not depend much on whether the low is only used for emergency or is used for low speeds as well.

The next two points are those to do with acceleration time for the compressor unit. Many people, particularly in America, have been worried about the lack of engine braking on the gas turbine and it is this point which helps us to solve the problem of acceleration. The idea is to have a shaft and gears with friction clutch and roller clutch to connect the power turbine shaft to the compressor turbine shaft. The connecting shaft has a gear ratio such that at about 30 m.p.h. in top gear the compressor unit would be running at 35,000 r.p.m. A roller clutch or free wheel is also provided so that the compressor unit can always overrun the car drive and run up to full r.p.m. A friction clutch, similar to a very small car clutch but capable of standing the required r.p.m. is controlled by one of several methods. Perhaps the simplest is a centrifugal arrangement connected to compressor speed such that the friction clutch is disengaged above something over two-thirds full compressor speed.

The operation is somewhat as follows, although there are various complications to watch regarding the exact timing of the various operations. A typical case is described since it is realized that this job can be arranged

in various ways. When the car is below a speed at which it would drive the compressor unit at its idling speed this ancillary equipment has no effect. If the car is being driven at high speed the roller clutch is free wheeling, because the compressor unit will be running very fast, and once again the equipment will have no effect. However, if the foot is lifted off the accelerator under these conditions, the compressor unit will try to slow to idling but the friction clutch will engage when the speed has dropped sufficiently and the compressor speed will remain high owing to the driving effect of the momentum of the car. This will cause a braking effect of an amount governed by the force which is placed on the friction clutch. Roughly the amount of braking would, in practice, be made about equal to third gear overrun on a normal car since this is suitable for holding the compressor speed reasonably high. Not only does this give engine braking but the acceleration time is also insignificantly short. In fact, acceleration of a compressor unit at the high speed end of its range is very rapid. It must be realized that the friction clutch will slip above a certain car speed, so that above this speed the torque and hence the car braking and compressor speed will be constant.

It will be perfectly obvious that this system suffers from the defect of being quite ineffective at low speeds. To overcome this necessitates a second complication. This is what has been termed electric acceleration, the first system being called regenerative braking although that is by no means a strictly correct term.

In this system the starter is connected to the engine through a free wheel or roller clutch, which is probably the best way of making a starter drive in any case, and controlled as follows. Two compressor speed sensitive switches are arranged in series in the starter lead, one arranged to cut the starter current at some speed just below idling and the other to cut at a speed just below that at which the friction clutch will drive the compressor. A contact on the accelerator pedal is arranged to close on full depression and is connected in parallel with the first pressure switch.

When the engine is started, if the accelerator is not fully depressed, the starter is cut out just before idling and simply slips the free wheel during running, but when the accelerator is fully depressed the starter is re-energized and aids the compressor until the higher speed is reached. Of course, it is necessary that the starter should be correctly geared but in practice this works out very well if the gearing is arranged so that peak starter power occurs at about idling speed. With this ratio, say 14:1, the first few revs of a start are rather sluggish, as the ratio is too high, and this gives time for the fuel pressure to build up, while the first part of the acceleration from idling, which is usually the most sluggish, has the maximum help.

If this electric acceleration is arranged suitably it can cut the acceleration time of the compressor to a quite short enough figure for the low car speeds. It is unsuitable for use without the regenerative braking as it would consume too much from the battery if used all the time, but in conjunction with the regenerative braking it would only be used when maximum acceleration was required at low speeds, that is, when the accelerator was fully depressed at a car speed insufficient to hold the compressor unit above the setting of the second pressure switch.

These arrangements may be quite unnecessary for acceleration as the engine may finally have quite adequate compressor acceleration without them, but it may be worth fitting the regenerative braking for braking on the engine and it would then be just as easy to use it for keeping the compressor speed up as well. Another aspect of this matter is that a small amount of fuel is saved, that is the amount required to accelerate the compressor to the speed at which it is held by the regenerative braking from idling. As for electric acceleration, seeing the starter equipment must be there in any case, very little extra is involved in using it to accelerate. Fig. 1 shows the change gear and reverse arrangement described together with the regenerative braking mechanism and a starter arrangement suitable for use with electric acceleration.

The car

One of the few popular suggestions about the difficulties of the gas turbine car that turns out to have any significance, is that of the disposal of exhaust gas, although this is not quite a question of a jet that will blow people off the road. Arguments put forward by the writer some time ago were to the effect that if a gas turbine was made with the same specific consumption as a piston engine then its heat rejected would be the same, that is, the turbine exhaust would have in it the same amount of heat and exhaust fumes as the exhaust of the piston engine plus the heat from the radiator cooling; and that the specific consumption should not be much less, or the gas turbine car would not be worth producing. While this is true as far as it goes, it fails to reach the important point that in the piston engine the radiator cooling air is blown out under the car and can meander out where it likes without creating any trouble as it has no fumes to leak into the car or annoy pedestrians. The exhaust, on the other hand, having highly concentrated fumes, must and can easily be blown clear of the car and being such a small jet is rarely caught fully by pedestrians.

In the turbine these two cannot be separated and it is impossible to make a car really fume proof. Actually, if petrol was used as a fuel this might be just possible with very good combustion, but because one of the advantages of the gas turbine is its ability to burn lower grade fuels, we must make a car capable of using them. Also, at present, combustion is not always good under all conditions.

For this reason the gas turbine exhaust contains more fumes than the piston engine exhaust and it has been found almost essential to discharge it upwards at the rear. It must be stressed that this is a fundamental point and the whole question of car design is based

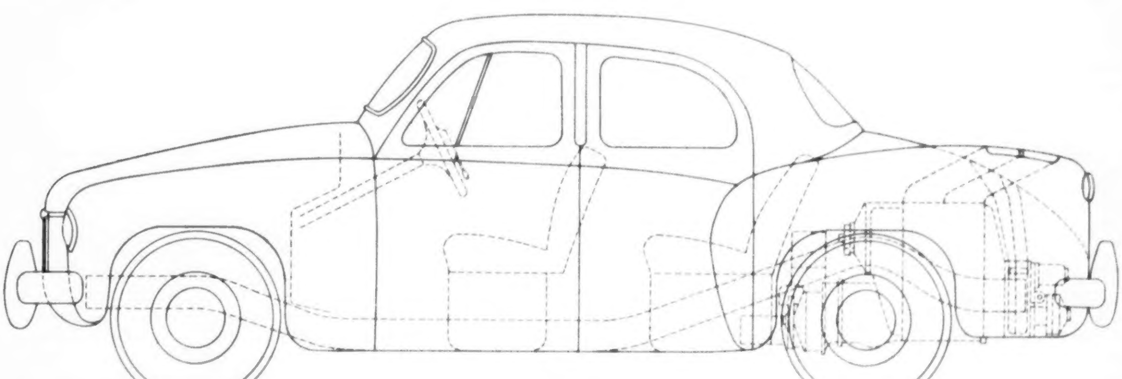


Fig. 5. 100 b.h.p. gas turbine car. Rear-mounted unit, central air intake, and twin exhausts in wings

on it since with the large ducting necessary for the turbine exhaust it has not been found feasible to take the exhausts up at the rear with a front engine without too much sacrifice of weight. If we accept that upward rear exhausts and therefore a rear engine becomes necessary, then we have, unfortunately, to enter a field of argument which has been covered at great length in the past without much in the way of positive conclusion and which the author hesitates to discuss. If possible it is, however, necessary to get clear ideas on the subject before proceeding further.

Briefly, it is generally recognized, although there may be some who will want to dispute this, that for a car to be really safe and satisfactory to drive, it must understeer. In other words, it must corner in such a way that any decrease in turning radius would be compensated by the car's automatically trying to reduce this decrease and so prevent an unstable tendency to get out of control. It is necessary for this effect to be instantaneous and not to require time if an unpleasant effect is to be avoided. This means that the only way of getting really good steering is to have a greater slip angle on the front tyres than on the rear. This requires either more weight on the front tyres or lower pressures in the front tyres, but there is unfortunately a limit to the amount which lower pressures can correct. Fortunately the gas turbine is very light, which would be a greater help if it were not for the fact that cars should be and to a certain extent are becoming lighter too. If the turbine is put at the rear it may possibly be necessary to have different rims for the tyres front and rear thus making the rear slip angle relatively less. It might be argued that some cars already have rear engines, but while these are satisfactory, they are not, in fact, amongst the best steering cars.

Another point to take into account is the question of aerodynamic stability. In a streamline body the centre of pressure tends to be well forward, and if the centre of pressure is ahead of the neutral steer axis the car is aerodynamically unstable and if swerved slightly at high speed will tend to swerve more and get out of control. The neutral steer axis is defined as that position where, if a force is applied to the side of a car the car will move squarely across the road without turning about this axis. This is, of course, with the car moving along the road. To summarize, for stability the centre of gravity must be ahead of the neutral steer axis in order to understeer and the centre of pressure behind the neutral steer axis for aerodynamic stability. It is very difficult to satisfy these conditions with the centre of gravity to the rear of the centre of the car as it is likely to be with the rear engine, and it is for this reason that rear engine cars are generally unstable.

However, two things can, if necessary, be done to improve this condition. First, from the aerodynamic point of view the car should be as low as possible at the front with a rather large area in profile at the rear either by merely having a large rear end or by a central fin or, as in a number of modern cars by bringing the rear wings upswept at the sides. This last is the most suitable as the exhausts, as will be described shortly, can be placed in them. As has been mentioned, if necessary the rear tyres may be put on wider rims or made larger to give less slip angle and so prevent oversteer.

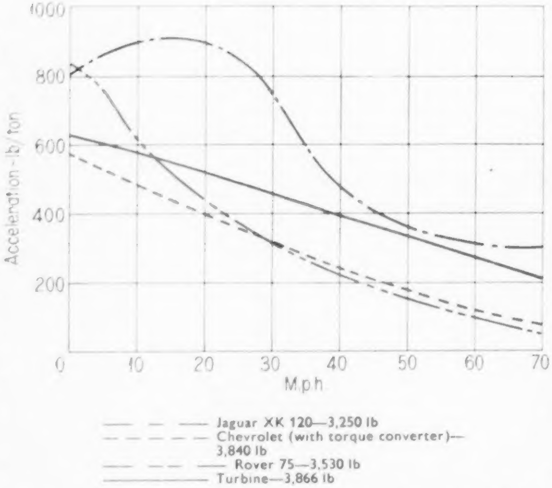


Fig. 6. Acceleration curves, calculated from *Autocar* Road Tests

If the engine is to be at the rear, it is necessary for a good mechanical layout to have independent rear suspension, which, for reasons too involved for this article, should be of the De Dion type. These difficulties of weight distribution may not be as bad as appears, since the Ordinary Rover 75 with the turbine placed in the rear is perfectly satisfactory. This may be because the ordinary Rover 75 is basically well designed and already has exceptionally good steering.

Silencing, which has often been suggested as a difficulty, is not really very much of a problem. First, it should be realized that there are really four different components to the noise one hears either outside or inside a car. There is the inlet noise, the exhaust noise, the noise emanating from the engine casing itself, and the noise transmitted through the mountings to the chassis and body. Dealing with these in turn, the inlet noise is easily reduced to a satisfactory level by simply providing a sound absorption or Burgess type silencer. This has to be about 18 inches long if it has no splitters but a single splitter or partition will make 9 to 12 inches quite adequate. It consists of a duct of perforated aluminium with about one inch of glass wool and an aluminium outer cover. The splitter is about one inch thick of similar perforated metal. The exhaust silencer is a very similar arrangement, except that as it must stand considerably higher temperatures, it should be made of aluminized mild steel with "Stillite" or other suitable high temperature sound absorber.

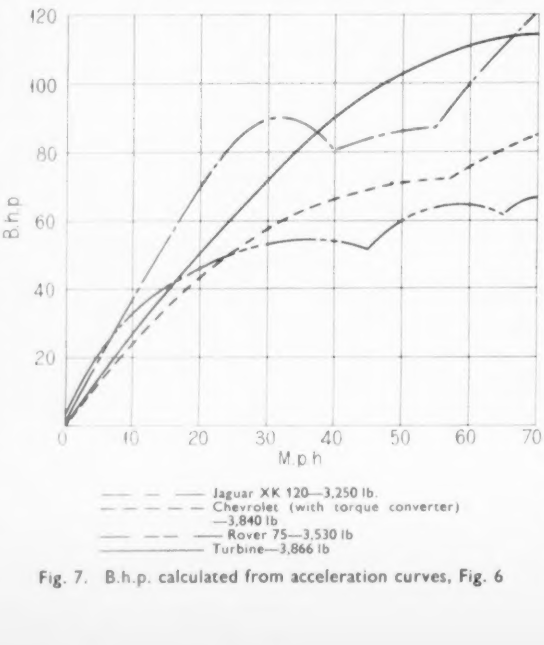


Fig. 7. B.h.p. calculated from acceleration curves, Fig. 6

Noises emanating from the engine casing is a function solely of the accuracy of balance and of the maintenance of balance. An engine with its compressor and power turbine shaft assemblies in accurate balance while running is extremely quiet but it has been found that some designs shift while running, a condition which obviously must be corrected. As for transmission of noise through the frame and body, the engine must be suitably rubber mounted but with a well balanced engine, which is essential in any case both for reducing direct noise and because bad balance will cause bearings to fail and the vibration cause breakages, this is quite an easy problem.

Fig. 5 shows a car designed on the lines suggested. The engine is as shown in Fig. 1

and is used in conjunction with a De Dion suspension. The intakes are in the top of the boot and the exhausts are from the top of the upswept wings. The styling is not meant to be taken too seriously as the design is only meant to give the general idea of what is required. The luggage would be under the bonnet.

In Fig. 6 the figures from the *Autocar* Road Tests have been converted into "Tapley Meter" readings in pounds per ton and are compared with those of a turbine car. These figures have been converted in Fig. 7 into b.h.p. at the road wheels. It can be clearly seen how the piston engine cars require three gears to produce their power at the lower speeds. Fig. 8 shows how a gas turbine car with good efficiencies can give as good a fuel consumption as a piston engined car provided it has a good heat exchanger.

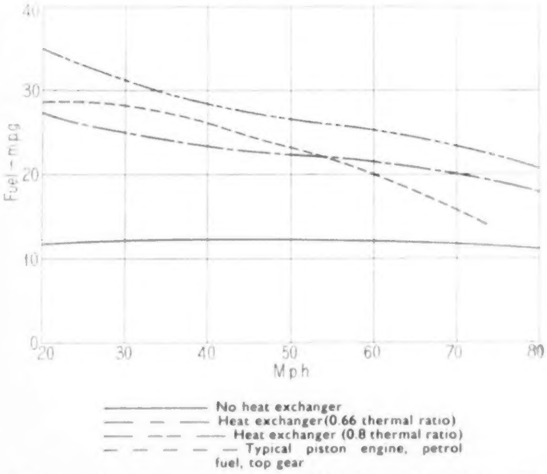


Fig. 8. Fuel consumption curves. 100 b.h.p. unit with turbine 85 per cent efficiency and compressor 78-75 per cent efficiency

Conclusions

It may appear that such a formidable list of difficulties and complicated solutions has been put forward as to constitute practically a proof of the impossibility of the gas turbine car ever getting into production. However, development up to the present by the Rover Company for an expenditure which is negligibly small compared with the development money spent on the piston engine, has gone a very long way to solving many of these problems; it has produced two cars which show very striking possibilities. It must also be remembered that many of the problems met in the development of the piston engined car looked almost insuperable in the early days, and if it had been realized that cars of to-day's standard were required many of the early pioneers may well have thrown in their hands.

It is estimated that the turbine car in production is likely to be as cheap or cheaper than the piston engined car, provided full production set-ups of blade machining is arranged on the right lines with advantage taken of the right techniques. The important point is to forget quickly anything learned from the aircraft industry who think of blade costs in pounds; and starting from scratch, to lower considerably the standards required and to think of blade costs in pence, coupled with the only system which can be cheap enough, i.e. machining the blades on the discs. This is primarily because the manufacture of small discs and blades is quite a different problem, but also because aircraft work is almost invariably done regardless of cost.

Apart from the blading and heat exchangers, the cost of the gas turbine is quite low, with any reasonable production set-up. The heat exchangers, like the blading, need the development of suitable techniques, but they are fundamentally a very suitable production job.

In servicing, it will probably be necessary to make suitable arrangements for quickly and easily replacing the heat exchangers or compressor unit and for sending the old compressor unit to a central depôt or the factory for reconditioning.

Having put forward the difficulties that beset the design and development of the turbine car, it is perhaps only fair to mention the advantages it would have when and if developed. First, it is extremely easy to drive, requiring no gear changing at all and having no jerks in its acceleration. Second, it will be quite free from vibration and noise and third, it should have a much longer life between overhauls and be much less sensitive to the type of fuel it burns. In practice, the aircraft gas turbine has shown itself to be much smoother and quieter with longer life and less fuel sensitivity, that is better in every respect.

Finally, it must be understood that this article is more in the nature of a summary of the problems and an interim report on their solutions. It clearly contains some highly controversial matter, which cannot be finally sorted out until development has reached a more advanced stage, and the opinions expressed are those of the author and not necessarily those of either the Rover Company or the *Automobile Engineer*.

KINEMATICS OF PISTON NOISE

A. MEIER, in an article in German in *A.T.Z.*, 1952, Vol. 54, No. 6, describes some research work carried out on piston slap, and suggests that one remedy which has proved of value is a slight lateral offsetting of the gudgeon pin from the piston centre without changing its vertical location. However, little information has hitherto been available on this method which is not well understood.

Tests by the Mahle KG on single and multi-cylinder four-stroke engines showed that without offset gudgeon pins, slap occurs only when a certain piston play, depending on the type of engine, is exceeded. With a fixed throttle setting, slap decreases with increasing rotational speed, and it is intensified by increased throttle opening. Moreover by offsetting the gudgeon pin, slap may be avoided even when the piston play is twice the nominal amount. To investigate these phenomena, a method of studying the cross movements of the piston was chosen based on using the piston and cylinder as the plates of a condenser. Variation in clearance changes the capacity of the condenser, and these changes, converted into variation in electric potential, can be rendered visible by means of an oscillograph.

Experiments were made on a single-cylinder test engine, pick-ups being arranged at the upper and lower ends of the piston skirt both on the major and minor thrust sides, to give a complete picture of the cross-movements of the piston. Calculations based on the dimensions and weight of the piston and connecting rod of the test engine gave good agreement with the experimental results. The mechanism of slap is explained by reference to results obtained with various offset pistons, oscillograms being given for the different positions. Measures to eliminate slap are essentially designed to minimize impairment of the oil film.

Ways of combating piston slap include: (1) lateral offsetting of the gudgeon pin by three per cent of the piston diameter towards the major thrust side; (2) lengthening of the piston skirt; (3) ensuring that the form rigidity of the piston is adequate to withstand operational stresses. Recommendations are also given on cylinder rigidity, gudgeon pin seating, methods of minimizing piston play, and on the engine cooling system which should ensure the lowest possible temperature difference between the upper and lower end of the cylinder (*M.I.R.A. Abstract No.* 6000.)

M.I.R.A. REPORTS

Recent Releases for General Circulation

A NUMBER of research reports have recently been released for general circulation. Among these is Report No. 1951/1 *The Experimental Determination of Static and Dynamic Stresses in a Double-Deck Public Service Vehicle.* The report describes a series of tests carried out on a London Transport Executive RT3 type double-deck public service vehicle. These tests were made to determine the stress distribution in various parts of the structure under conditions of both static and dynamic loading. Wire resistance strain gauges with the appropriate equipment were used to make strain measurements on the chassis frame, body vertical pillars, transverse floor members, rear platform supports and external body panels.

In the static load tests, the stresses due to the weight of the major components were found by dismantling the vehicle in stages. In addition, the stresses were measured in a complete vehicle with a passenger load of 8,200 lb. The maximum total static stress measured in the frame was 3·5 ton/in^2. This stress, which was compressive, occurred in the upper flange of the nearside channel member. Stresses in the body structure were, in general, considerably lower. However, a maximum of 4·45 ton/in^2 compression was measured in a vertical pillar just below its abutment with one of the hoopsticks carrying the upper saloon load.

Dynamic tests included bumping, cornering and brake tests. The bumping tests consisted in running over a 2 in board at 20 m.p.h. with all four wheels, and again with the nearside wheels only. A maximum stress of 5 ton/in^2 compression was measured in the chassis frame and 4·25 ton/in^2 compression in the body structure, both stresses occurring at the same points as those already mentioned in the static tests. Further experiments showed that stresses due to bumping were approximately proportional to the height of the bump, but with the 2 in bump they did not increase with speed above approximately 20 m.p.h. The stresses were of an oscillatory nature with a frequency of about 7·1 cycles/sec. This appeared to be a natural frequency of vibration of the frame.

The maximum stress in the chassis frame due to cornering was 6 ton/in^2 tension. It occurred near the attachment of the rear stabilizer. In the body, an unusually high value of 7·9 ton/in^2 compression was measured in one of the transverse flitch beams, by means of which the body is supported on the chassis. Stresses due to braking were generally the lowest encountered, the maxima for the chassis frame and the body being 2·5 ton/in^2 tension and 2·2 ton/in^2 tension. They occurred respectively in the upper flange of the side member and in the transverse flitch beam.

Apart from obtaining the stress values, the tests also served to further the technique of this type of investigation, and to examine the suitability of the four-channel dynamic strain gauge recording equipment described in Report 1949/5. Development of this equipment had been based on experience with single-channel recording apparatus in tests on a large passenger car and a commercial vehicle.

Report 1951/7 is entitled *Further Experiments on Noise in Vehicles.* It describes how detailed octave-band analyses were made in three saloon cars, in order to investigate the correlation between objective measurements and subjective estimates of "noisiness" in vehicles. Three cars, each of a different make, were chosen for the tests which were made at speeds ranging from 20 to 55 m.p.h. The noise characteristics of the cars are discussed with reference to differences in their mechanical construction.

Detailed objective octave - band analyses of the type outlined in the report are, it is suggested, capable of correct subjective interpretation, and define adequately the "noisiness" of vehicles. The analyses show objective differences in "noisiness" between cars, and subjective estimates are in general agreement.

Report 1951/8, *The Failure of Gears by Pitting*, is a companion to Report 1950/7, which dealt with the bending fatigue strength of the teeth of gears made from the same steels as are covered in the later report. It gives the results of an investigation into the relative resistances to pitting, of gears made from seven typical alloy gear steels. These steels are:

En 24 (1½ per cent Ni-Cr-Mo oil hardening), both plain and leaded.
En 30A (4¼ per cent Ni-Cr air hardening).
En 36 (3 per cent Ni-Cr case hardening).
En 39A (4¼ per cent Ni-Cr case hardening), both plain and leaded.
En 40C (3½ per cent Cr-Mo-V nitriding).

Gear pairs made from these steels were tested in the 5 in centres, power-circulating gear testing machine (I.A.E. Report 1944/9) at various loads, and the materials were compared on the basis of their S-N curves, where S is the surface stress criterion S_c, commonly used for evaluating the surface loading of gears, and N is the number of stress cycles necessary to cause pitting. The results confirm that pitting is a form of surface fatigue failure, the life varying inversely with the stress applied. However, over the range of life covered, that is 10^5-10^7 cycles, the S-N curves gave no indication of the presence of a fatigue limit below which failure would not occur.

Resistance to pitting appears to depend upon the surface hardness of the materials. The three case-hardened steels had a load carrying capacity about 70 per cent higher, for any given life, than the three through-hardened steels. A still greater resistance to pitting at all except the higher applied loads was evidenced by the nitrided steel. The use of lead in En 24, through-hardened steel, to improve machinability, gave no reduction in pitting strength, but in En 39A case-hardened steel, it caused a reduction of about 15 per cent in the load carrying capacity.

Report 1951/11 is entitled *Development of a Test Procedure for Lubricating Oils for British Road Vehicle C.I. Engines.* At the invitation of several member firms, M.I.R.A. investigated engine test procedures, using a six-cylinder C.I. engine of a road vehicle for judging the performance of lubricating oils on a "go" or "not go" basis within 50 hours. The criterion used to distinguish between the oils is the extent to which the piston rings stick. This is determined not only by inspecting the pistons at the conclusion of the test, but also by observing the variation in the blow-by during the course of the test.

Two oils were used in the investigation, and two fuels were employed, one with 0·2 per cent and the other with 1·0 per cent sulphur content. For the purpose of the tests, samples of the oil were each blended with different detergent additives, and further samples were used plain. Three detergent oils commercially available in 1948 were used in the investigation. With the low sulphur fuel all the additive oils passed the test, but both the plain oils failed. However, with the higher sulphur fuel, only two of the additive oils passed, and the plain oils again failed.

Under the auspices of the Institute of Petroleum, the same oils were tested by the C.R.C. L-1545 (Caterpillar) procedure, using low sulphur fuel, in the laboratories of several oil companies. The same distinction between plain and additive oils was observed. Concurrently with the laboratory tests, two bus companies carried out extensive service tests on these oils. The tests, which involved short stage bus operation, showed little if any improvement due to the use of additive oils. Thus, neither the laboratory tests so far carried out by M.I.R.A. nor the Caterpillar L-1 test can be regarded as applicable to this type of service. Possible reasons for this lack of agreement are considered, as well as the feasibility of correlation with other types of service. It is stated that there appears to be a need for further research, and suggestions are made as to the lines on which work should be carried out.

STRUCTURAL DESIGN

Part I: An Analytical Method for Chassisless Vehicle Design

By T. K. Garrett, A.M.I.Mech.E., A.F.R.Ae.S.

In the February issue of Automobile Engineer, *a method of calculating the maximum loads that may be imposed on a motor vehicle structure was described. However, a knowledge of the loads is, by itself, only of academic interest. Such knowledge, to be of practical use, must be applied to design a structure capable of withstanding those loads.*

In this article, the loads are applied to a vehicle structure and their path through that structure is traced. By this means a careful check may be made before the prototype is built, to ensure that every component and every welded joint is capable of performing the duty required of it. Although subsequent development is still necessary, the amount of modification required is reduced, and difficulties arising from the need to alter tools are to a large extent obviated. The subject will be dealt with in two sections. This, the first, deals with the initial design work and the rear end structure.

IN the early days of the motor car, the body was regarded only as a means of sheltering the passengers from the weather and dust; and the chassis was self-contained so far as strength was concerned. However, as the years have passed, it has been increasingly realized that a suitably designed body can take a large proportion of the loads. Indeed, very little reinforcement is required to make it capable of being completely self-sufficient in this respect.

These discoveries have not been made fortuitously. They have been occasioned by the pressing need for economy in space and materials to meet increasing industrial competition throughout the world. If a chassis frame is to be strong enough by itself to carry all the loads, it must be relatively heavy. Moreover, the steel that goes into the body is then redundant from the point of view of strength. Of recent years, steel has been in short supply, and this has further encouraged the trend towards chassisless construction.

Development has taken place in two stages. The first was the introduction of *unitary construction*, in which a more or less conventional underframe was welded to the body. Because of its positive attachment to the body, this frame could be made of sheet steel of a lighter gauge than was possible in a separate frame. More recently, *chassisless construction* has been introduced. In this there is no underframe at all in the generally accepted sense of the term. Two good examples of this type are the latest Austin 7 and the S.A.A.B. 92.

The lighter forms of construction have other advantages besides economy of materials. Experience has shown fairly conclusively that the total cost of a vehicle varies in proportion to the weight. Thus, by reducing weight, other economies are automatically introduced at the same time. Another advantage of light construction is an improvement in performance, and fuel economy.

One of the principal disadvantages of the lighter forms of structure is that the ratio of surface area to volume is large by comparison with the earlier, more solidly built vehicles. This increases vulnerability to corrosion. However, it is claimed that the improved anti-corrosive treatments, such as bonderizing and dipping in paint, now coming into use have more than offset this disadvantage.

The idea that chassisless vehicles are not as strong or as stiff as the more conventional ones is a fallacy. The relatively large depth of body panels used as beams ensures that both the strength and stiffness are greater than in the case of a chassis frame, provided these panels are suitably stiffened and supported to prevent buckling. Indeed, the stiffness of the chassisless vehicle is an important factor contributing to its good riding qualities. It has been shown, as for example in the case of the Austin 7, described in the December, 1952, issue of *Automobile Engineer*, that very little additional reinforcement to the body is needed to make it structurally self-sufficient.

It is debatable as to which form of construction is the easier to repair. If a vehicle has been damaged to such a degree that a new body is required, it is likely that the frame will also have to be replaced. But in any case, if a new body is required, it would not appear to matter much whether it is a chassisless body or a more conventional one. With regard to lesser damage, detachable wings, and spare assemblies, as well as spare components, would probably meet most requirements.

Chassisless structures are much more complex than a simple chassis frame. Because of this, it is easier to make mistakes in the design. Another important difference between these two forms of structure is that in one type, the components must be mounted on relatively thin gauge metal, which must be shaped or supported locally so as to be stiff enough to carry them; while in the other type, the thicker gauge of the chassis frame, and its compact box or channel sections, make the attachment of such components as the steering box, and suspension bearings easier.

It is clear, therefore, that more analytical thought than has hitherto been necessary must be devoted to design. The loads that the structure must take need to be carefully analyzed, and their path through the structure traced so that, before the vehicle is made, it may be determined whether each component is strong enough. It is only in this way that it is possible in such a complex structure as a motor body, to ensure that there is a satisfactory path through which the loads may travel.

Stress analysis

For the purpose of demonstrating how the analysis may be performed, the system of loads described in "Automobile Dynamic Loads" in the February, 1953, issue of *Automobile Engineer* will be applied. An imaginary structure has been devised for the demonstration, Fig. 1. It incorporates features that may be found in several modern chassisless designs.

The first stage in the design process is to estimate the bending moments and shearing forces on the laden vehicle under a vertical acceleration force of 4·5 g. When the necessary calculations are completed, the appropriate diagrams are drawn, Fig. 2. From these, the dimensions of the body sill may be determined, and an approximate idea obtained of the proportions needed for the remainder of the structure.

For the purpose of making the calculations, the following weights will be assumed: Vehicle kerb weight, 1350 lb; unsprung, 200 lb; engine and gearbox, 250 lb; passengers, 200 lb each. Luggage carried aft of the wheels would relieve the bending moments between them; therefore the diagram will be plotted first for the case with no luggage, and then for the design of the rear end only, with 112 lb of luggage in the boot. The centre of gravity of the engine is estimated to be directly over the axis of the front wheels, while the centre of gravity of a man in a sitting position is located approximately on the front wall of his stomach at waist level. The sprung weight, minus that of the engine, passengers and luggage, is assumed to be uniformly distributed over the 130 in length of the vehicle.

Thus, the uniformly distributed load, or UDL:

$$= \frac{4{\cdot}5\,(1350 - 200 - 250)}{130}$$

$$= 31{\cdot}2 \text{ lb/in.}$$

The reaction R_F at the front engine mounting is obtained

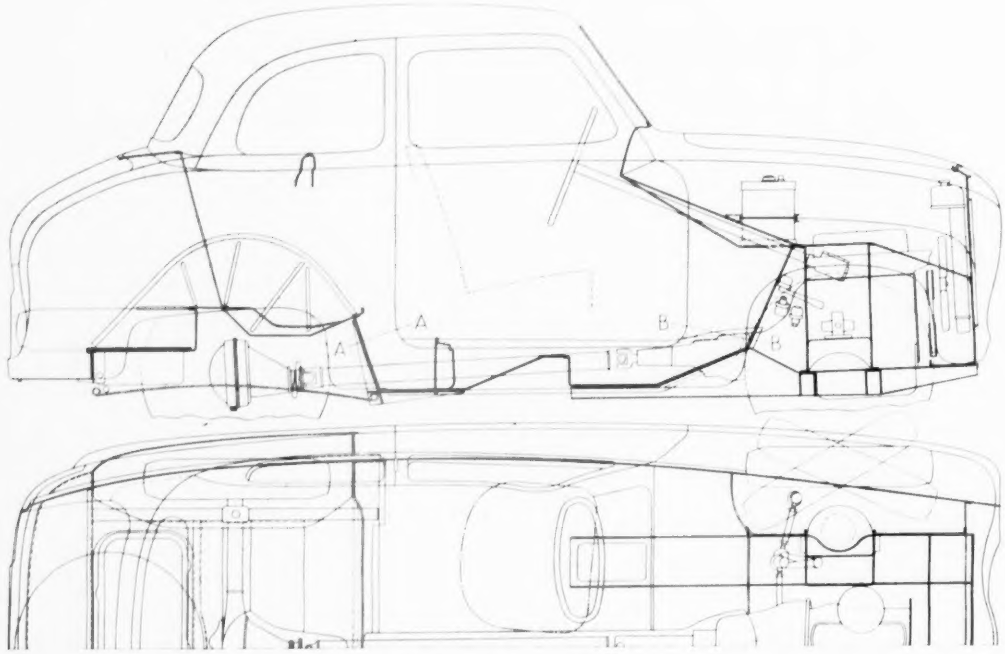


Fig. 1. The vehicle structure

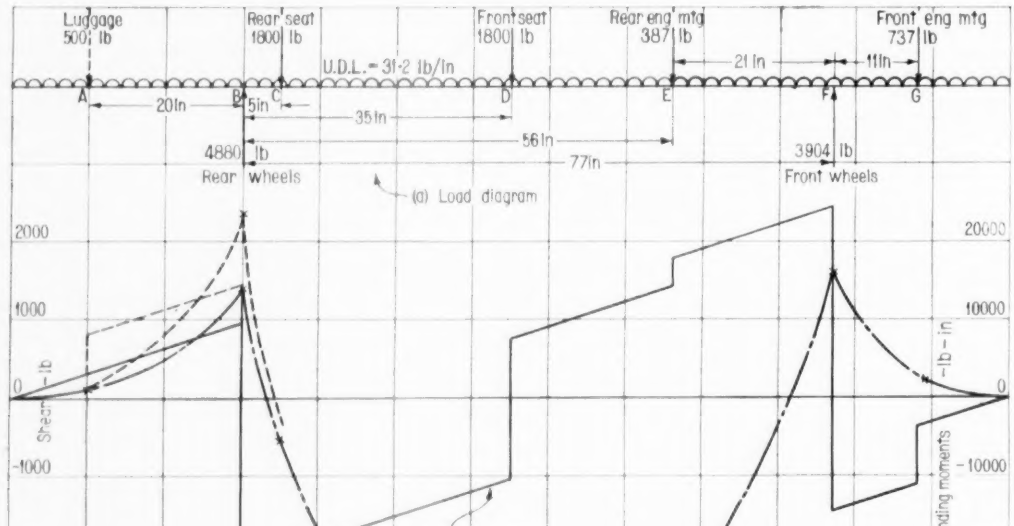


Fig. 2a. Load diagram
b. Shear force diagram
c. Bending moment diagram

The dotted curves at the rear (left-hand side) are shears and bending moments with the boot loaded

by taking moments about the rear mounting:

$$R_F = 4{\cdot}5\left(\frac{250 \times 21}{130}\right)$$
$$= 737 \text{ lb}$$

and the reaction at the rear:

$$= 4{\cdot}5(250 - 164)$$
$$= 387 \text{ lb.}$$

Taking moments about the rear wheels for the front wheel reaction, we have:

$$\frac{(31{\cdot}2 \times 130 \times 35) + (1800 \times 5) + (1800 \times 35) + (737 + 387)}{77}$$
$$= 3904 \text{ lb.}$$

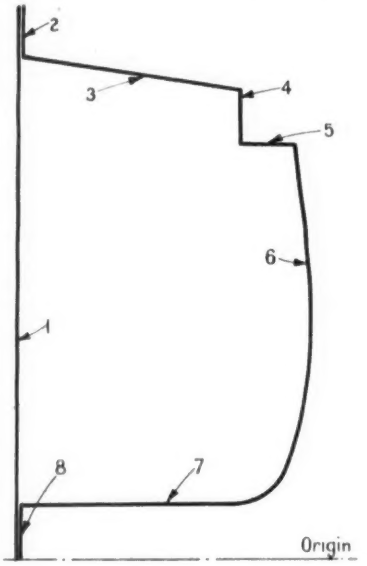


Fig. 3. Sill section

It will be noticed that a certain amount of arithmetical work has been saved by the fact that the engine happens to be directly over the front wheels.

The rear wheel reaction is:

$$[(31{\cdot}2 \times 130) + (2 \times 1800) + 1124] - 3904\text{lb}$$
$$= 4880 \text{ lb.}$$

This figure should be checked by taking moments about the front wheels.

From the above data, the load diagram is completed, Fig. 2a. Next, the shear forces are computed and the diagram drawn, Fig. 2b. Then the calculations for the bending moment diagram, Fig. 2c, are made. These are best done in two parts, that is to say, by working first from one end towards the centre and then from the other to the centre. This makes the calculations less tedious. It need hardly be added that if either the bending moment or the shear force diagram does not close, there is an error in the calculations. In general, errors of up to 2 per cent are regarded as acceptable, because manufacturing tolerances, etc., will result in variations in the strength of components of at least as much as this. Moreover, the loads are not accurate within such close limits.

Bending moments

A.	$31{\cdot}2 \times 10^2 \times 0{\cdot}5$	$= +1560$
B.	$31{\cdot}2 \times 30^2 \times 0{\cdot}5$	$= +14050$
C.	$31{\cdot}2 \times 35^2 \times 0{\cdot}5 - 4880 \times 5$	$= -5300$
D.	$31{\cdot}2 \times 65^2 \times 0{\cdot}5 - 4880 \times 35 + 1800 \times 30$	$= -51100$
G.	$31{\cdot}2 \times 12^2 \times 0{\cdot}5$	$= +2250$
F.	$31{\cdot}2 \times 23^2 \times 0{\cdot}5 + 737 \times 11$	$= +16350$
E.	$31{\cdot}2 \times 44^2 \times 0{\cdot}5 + 737 \times 32 - 3904 \times 21$	$= -28200$
D.	$31{\cdot}2 \times 65^2 \times 0{\cdot}5 + 737 \times 53 - 3904 \times 42 + 387 \times 21$	$= -50980$
		-51100
	Total error	$= 120$
	Percentage error	$= 0{\cdot}235$

The sill

From Figs. 1 and 2, it can be seen that the maximum shear for which the sill must be designed is about 1,900 lb/vehicle or 950 lb/side, while the maximum bending moment in this component is 51,100 lb-in/vehicle. A sill section 5 in deep is proposed, Fig. 3. Since the sill has two side walls, the shear distribution is 475 lb/in, which, when divided by the gauge of the metal (0·036) and 2240 gives a stress of 0·59 tons/in^2. This is obviously well below the shear strength of the material.

In order to check the bending strength of the sill, it is necessary to calculate its moment of inertia. A very convenient way of doing this for complicated sections is given in the table at the foot of this page.

The first column refers to those portions of the section labelled in Fig. 3; their sizes, which are given in column 2, are used for calculating both the area and the local I, $bd^3/12$, given in columns 3 and 8 respectively. It will be noticed that figures for two of the items have been omitted because inspection of Fig. 3 shows that the moment of inertia I about their neutral axis will be a negligible quantity relative to the total. The symbol "y" refers to the distance, in inches, of the centre of area of the item under consideration from the arbitrary origin. This origin may be any line parallel to the neutral axis about which the bending moments are applied. The amount of work necessary to complete the calculations may, of course, be reduced to a minimum by a suitable choice of position for the origin. Having completed all the columns and summed the last three and the area column, the calculations are continued as follows:

Let $\bar{y}_B$ = the distance from the neutral axis to the lowest extreme fibre, then $\bar{y}_B = \frac{Ay}{A} = \frac{1{\cdot}252}{0{\cdot}513} = 2{\cdot}44$ in, and $\bar{y}_T$, or the distance from the neutral axis to the uppermost extreme fibre $= 5 - 2{\cdot}44 = 2{\cdot}56$ in.

Then, by the theorem of parallel axes:

$$I = Ay^2 + \text{local } I - A\bar{y}^2$$
$$= 3{\cdot}896 + 0{\cdot}502 - 0{\cdot}513 \times 2{\cdot}44^2$$
$$= 1{\cdot}348 \text{ in}^4$$

$$\text{and } Z_T = \frac{1{\cdot}348}{2{\cdot}56}$$
$$= 0{\cdot}525 \text{ in}^3$$

$$\text{while } Z_B = \frac{1{\cdot}348}{2{\cdot}44}$$
$$= 0{\cdot}552 \text{ in}^3$$

where Z_T and Z_B are the two moduli of section.

The bending moment/sill is:

$$\frac{51100}{2} \text{ lb-in, so that the highest stress}$$
$$= \frac{51100}{2 \times 0{\cdot}525 \times 2240}$$
$$= 20{\cdot}7 \text{ ton/in}^2.$$

This is on the assumption that the sill takes all the bending loads. In cars such as this, in which there is a propeller shaft tunnel adequately supported at the dash and heelboard, the tunnel will usually relieve the body sills of some of the bending moments due to the weight of the passengers. Moreover, in saloon cars, the roof will also help. However, it is probably a good thing to make the sill self-sufficient so far as bending is concerned. Then a convertible version may be made from the same tools.

Steel sheet generally used in bodywork has a yield stress of about 16 ton/in^2 and an ultimate of 20 ton/in^2.

Item	Size	Area	y	y^2	Ay	Ay^2	Local I
1	0·036 × 5·0	0·180	2·5	6·25	0·4500	1·1250	0·374
2	0·036 × 0·5	0·018	4·75	22·5	0·0855	0·4050	0·0004
3	0·036 × 2·0	0·072	4·4	19·36	0·3170	1·3950	—
4	0·036 × 0·5	0·018	4·0	16·00	0·0720	0·2880	0·0004
5	0·036 × 0·5	0·018	3·75	14·05	0·0675	0·2530	—
6	0·036 × 3·25	0·117	1·875	3·52	0·2190	0·4120	0·103
7	0·036 × 2·0	0·072	0·5	0·25	0·0360	0·0180	0·024
8	0·036 × 0·5	0·018	0·25	0·0625	0·0045	0·0001	0·0004
		$\Sigma A = 0{\cdot}513$		$\Sigma =$	1·252	3·896	0·502

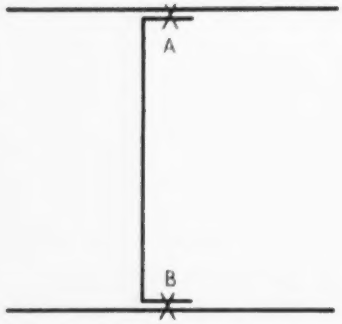


Fig. 4. The flanges A and B of the vertical beam section are fully stabilized by the top and bottom panels

Nevertheless, for the purpose of this article it will be assumed that the stress, calculated to be 20·7 ton/in^2, will be satisfactory because of the relieving factors just mentioned. However, should it be required at a later date to make a convertible version, it must be borne in mind that it may be necessary to increase the thickness at least of the inner plate, and possibly of the whole sill section, to 18 s.w.g. An increase to 18 s.w.g. would increase the strength by approximately $0{\cdot}048/0{\cdot}036 = 1{\cdot}33$, and the stress would be decreased to 15·5 ton/in^2.

A study of the structure, Fig. 1, will show that to the rear of the sill, the bending loads are taken by the outer quarter panel, and by an inner quarter panel attached to the sill inner plate and rear pillar. This inner panel extends over the rear wheelarch to which it is spot welded, and its upper edge is stabilized beneath the rear quarter light by its attachment to the outer panel as well as by its shape, while to the rear it is stabilized by its attachments to the side of the parcel shelf and to the drip channel in the boot opening. In view of the unstable condition of the lower edge of the outer quarter panel, it will be assumed that all the load is taken by the inner one.

The slenderest section of the inner quarter panel is immediately behind the foot of the rear pillar. At this point,

Fig. 6a. The true stress distribution diagram when the extreme fibre stresses exceed the yield stress in a beam under a bending load
b. The stress distribution assumed for stress calculations based on the commonly used theory

the section AA is 5 in deep, and the bending moment/vehicle is 33,000 lb-in. This bending moment will induce tensile and compressive end loads of approximately $\frac{33000}{2 \times 5} = 3300$ lb/side, and if the stress is to be limited to 12 ton/in^2, the area of metal required to take that compressive end load is $\frac{3300}{12 \times 2240} = 0{\cdot}123$ in^2, which may be provided by 3·4 in of 20 s.w.g. steel. The figure of 12 ton/in^2 has been found by experience to be the maximum safe compressive load in mild steel structure fabricated from thin sheet. Stresses in excess of this are liable to cause buckling. Moreover, stresses appreciably lower than 12 ton/in^2 may cause buckling failure unless the metal is fully stabilized in two planes, as in Fig. 4. In the illustration, the vertical beam section AB is spot welded between two large flat sheets of metal so that under compressive end loads at either A or B buckling of the flanges cannot occur in either the vertical or the horizontal plane until the stress exceeds a value of approximately 12 ton/in^2.

Experience has further shown that a thin sheet of metal is stable, that is to say, it will not buckle or form waves for a distance of 16 t from its point of stabilization, where "t" is the thickness of the metal. For example, the effective width of metal at A, Fig. 4, capable of carrying 12 ton/in^2 is 16 t to the left of the spot weld represented by an X, and 16 t to the right of it, plus 16 t of the flange of the beam (assuming the flange is large enough), plus 16 t of the web immediately below the flange. Stabilization of the web at this point, of course, is effected by the right-angle bend of the flange. In 20 s.w.g., 16 t is 0·575 in, the total area capable of carrying this load at A or B is therefore:

$$0{\cdot}575 \times 4 \times 0{\cdot}036 = 0{\cdot}0827 \text{ in}^2.$$

It can be seen that the section at the junction of the inner quarter panel with the heelboard on section AA is similar to A in Fig. 5, so that if there were only a single thickness between A and B, there would be $4 \times 0{\cdot}575 = 2{\cdot}3$ in of 20 s.w.g. available to take the load. This deficiency may be remedied easily by extending the sill inner panel upwards and the inner quarter panel downwards so that they overlap as in the illustration, thereby doubling the strength of the web portion. At the rear, one of the two panels may be flanged inwards and the other outwards, so that both may be spot welded to the heelboard. The sill outer panel will have to be assembled after this operation.

A word of warning is necessary. This approximate method of determining stresses by estimating them from the end load and area of metal capable of taking the load is only reasonably accurate when the section is symmetrical. If the neutral axis is displaced an appreciable amount from the centre of the section, the stress can be much larger in one boom than in the other. In such cases, the tabular method, as used for determining the modulus of section of the sill, must be employed. The section AB in Fig. 5 is nearly symmetrical because the rear pillar cannot be assumed to be fully effective as it is more than 16 t in depth.

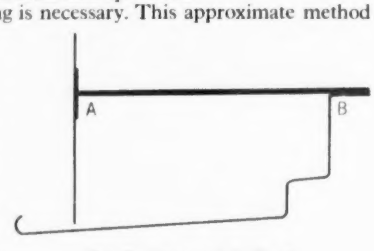


Fig. 5. Section AA, Fig. 1

Sometimes, checking the area of metal required to take the end load will show the section to be slightly under strength. Then the more laborious tabular method may be used, allowance being made for any parts of the section that are unstable. As calculated from this moment of inertia, the stresses are slightly lower than those obtained by the approximate method used for the quarter panel section. In addition, by applying a suitable *form factor*, a beam can be shown to be capable of carrying heavier loads than would appear possible from the simple engineers' bending theory, in which the stress is given by M/Z. The theory of form factors is described in Scientific and Technical Memoranda No. 5/45, of the Ministry of Aircraft Production, entitled *Elements of Stress Analysis*.

A form factor is the ratio of the load that a beam will carry to the ultimate load as calculated from the M/Z

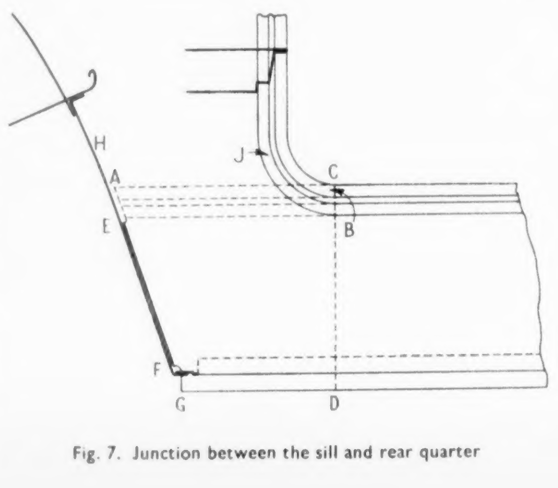


Fig. 7. Junction between the sill and rear quarter

formula. This formula is based on the assumption that the stress distribution over the depth of a beam is as in Fig. 6b; this assumption holds good provided the stress in the extreme fibres is lower than the elastic limit. However, when these extreme fibres are subjected to higher stresses they yield, and extra load is taken by fibres nearer the neutral axis, so that the stress distribution is then as in Fig. 6a.

Another condition under which the simple bending theory does not hold good is where the length of the beam is small relative to its depth. This is because the stress distribution is much more complex than that on which the bending theory is based. It has been found by experience that for a cantilever whose length to depth ratio is 1 : 2, and for a simply supported beam where the ratio is 1 : 1, the strength is $1\frac{1}{2}$ times that indicated by calculation.

An often overlooked and most important feature of the design of motor vehicles is the provision of an adequate structural path for the transfer of loads from one component to another. There must be a continuous structure for the transfer of both the shear and bending moments between the sill and the front and rear end of the vehicle. In the case of the rear end, the sill outer panel is joggled under the rear quarter outer panel and there is a vertical soldered joint, CD Fig. 7, below the point where the radius at the lower rear corner of the door opening joins the horizontal upper surface of the sill. At the extreme rear end, the sill is closed by flanging outwards its outer panel and spot welding it to the heelboard, at EF.

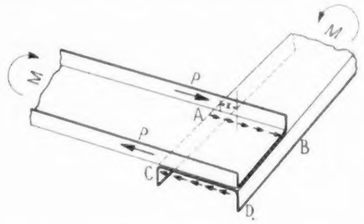


Fig. 9. A method of obviating the need for a kink strut

The depth of the section is five inches and the bending moment 16500 lb-in/side, so that the end load to be transferred from the top and bottom of the section to the inner quarter panel is 16500/5 = 3300 lb per flange. The following are some test results, giving the shear strength per spot of well executed spot welds joining two thicknesses of sheet metal:

s.w.g.	En 2A
10	3900 lb
12	3100 lb
14	2500 lb
16	2000 lb
17	1700 lb
18	1500 lb
20	1100 lb
22	800 lb

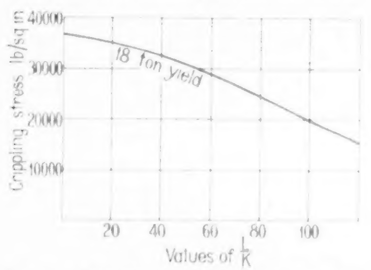


Fig. 11. Crippling stress for a tubular steel strut with pinned ends

Because of the possibility of inconsistencies in spot welding, it is advisable to apply to the above results a factor of safety of at least four. Some authorities consider that a further reduction of about 60 per cent should be applied when estimating the strength of welded joints. It follows that less than four spot welds should never be relied on to carry even a light load, because the employment of a factor of safety of four could be taken to imply that three out of every four spot welds are completely ineffective.

It can be seen that, with 20 s.w.g. panels, at least twelve and possibly twenty spots are needed to transfer the 3300 lb end loads from the sill to the inner quarter panel. These spot welds will have to be in rows of twelve or more, one row adjacent to the door pillar, and another at the heel board. Similar rows will be required at the top and bottom horizontal flanges of the sill. The load at the rear edge will pass from the sill inner panel into the heelboard, and thence into the inner quarter panel, while that at the front will go into the rear pillar section. About half the load will be transferred directly from the sill outer panel to the inner quarter at the top and bottom horizontal spot welded flanges of the sill at AB and GD.

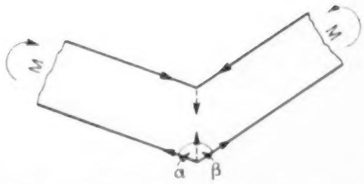


Fig. 8. Kink loads, indicated by dotted arrows, are induced at a junction between two straight beams

The junction of the sill with the other portions of the structure at the front and rear ends is a position where application of the correct structural principle is important, but is often ignored, with unfortunate results. It is the principle that must be applied to any beam shaped similarly to that in Fig. 8. When loaded with the couples "M," compressive end loads are induced in the top fibres of the beam, and tensile loads in the lower ones. Thus, the flanges at the bend tend to be forced, as indicated by the dotted arrows, towards the neutral axis of the beam. This tendency must be reacted by means of a component, usually termed a *kink strut*, otherwise the sheet metal of the web will alternately buckle and straighten as the load varies, and fatigue cracks will result. A kink strut should, of course, support the flanges as well as the web to which it is welded. The load in the strut will be the end load in the flanges at the kink multiplied by ($\cos\alpha + \cos\beta$).

An alternative is the method employed in Fig. 9, where two channel sections are welded together back to back. In this case, the end loads are transferred in shear from the flanges of one into the web of the other. If the end load at A and C is P, and the lengths AB and CD are each L, then the shear to be transferred is P/L lb/in. Thus at any distance x from A or C, the end load in the flange is reduced to $P\left(\frac{L-x}{L}\right)$, so that the end loads are zero at B, D and C. For this reason, the flanges may be tapered down on the top channel towards B and D, and on the bottom one towards C and D. The safe load, that is the test load reduced by a factor of four, is about 1000 lb/in for gas welded sheets of 20 s.w.g. mild steel. If spot welds are employed, they must be pitched closely enough to take off the requisite load per inch.

The incorporation of a kink strut at the junction between the sill and the rear quarter panel is rendered difficult by the relatively large fillet radius at the lower rear corner of the door. This same radius also makes it difficult from the practical point of view to complete the structure in the manner indicated in Fig. 9.

To meet structural requirements, a satisfactory arrangement might possibly be made by pressing the rear end of the sill so that its upper face fits around the radius and then extends horizontally between H and J, Fig. 7. Then the radius could be supported by a kink strut of top hat section between B and the lower corner FG. However, this would entail a certain amount of waste metal in cutting the blank for the sill outer panel.

In these circumstances, it might be better to maintain a straight sill section and weld in a top hat section strut vertically below B, Fig. 7, to complete the structure as in Fig. 9. The flanges of the top hat section should be

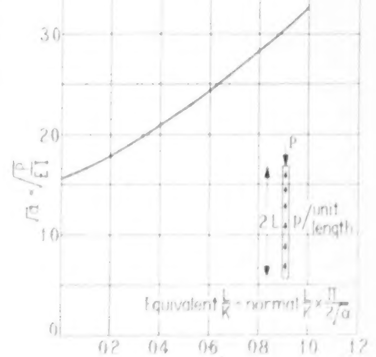


Fig. 10. Curve for determining the equivalent L/K of a strut with varying end load

spot welded to the sill inner plate and the section should be so positioned that the upper surface of the sill outer panel rests on its upper end. This is far from ideal, since it leaves an unsupported portion of flat plate between the upper face of the sill outer panel and J, Fig. 7. However, subsequent testing of the prototype will show whether or not further steps are needed to strengthen it.

At its upper end, the kink strut must be capable of supporting the 3300 lb compression load, which must be transferred to the sill inner panel in shear by at least 12 spot welds. The strength of a strut such as this, in which the end load varies along its length, is found as follows. First its equivalent L/K ratio is found where L = length and K = radius of gyration. This equivalent is obtained by multiplying the L/K, calculated in the usual manner, by $\pi/2\sqrt{\alpha}$. The value of $\sqrt{\alpha}$ may be determined from the curve Fig. 10, and K is either obtained from an engineers' reference book giving K for the more common sections or it is calculated from the expression $K=\sqrt{I/A}$, where I is the moment of inertia and A is the cross-sectional area. Next, the crippling stress may be estimated from strut curves such as Fig. 11. Where L/K is more than 100, the Euler crippling load should be calculated. It will be realized, of course, that over a long period of time a series of strut curves and other design data sheets may be drawn up and verified by tests. These curves will save a great deal of time.

The front end of the sill must be designed in a similar manner to the rear end. However, the task is much easier, because the minimum depth of section, BB on the dash side, is 8 in, so that the stresses are considerably less severe. There will not be space in this article to do the calculations for every part of the vehicle, so attention will now be given to the main structure around the front and rear suspension.

Rear end structure

The design loads for the rear as well as the front of the vehicle will be given by the design cases described in "Automobile Dynamic Loads," published in the February, 1953, issue of *Automobile Engineer*. Case 3, hitting a bump while accelerating, and Case 4, hitting a bump while cornering, are likely to be critical for the rear end, Fig. 12. The symbols used in the formulæ may be evaluated as follows:

R	$=\dfrac{4850}{2\times4{\cdot}5}+\dfrac{112\times97}{2\times77}$ $=610$ lb	T	$=43$ in
		ψ	$=78$ deg
		ϕ	$=24$ deg
θ	$=48$ deg	Cos ψ	$=0{\cdot}208$
Tan θ	$=0{\cdot}743$	Sin ϕ	$=0{\cdot}407$
W	$=2060$ lb	Tan ϕ	$=0{\cdot}445$
$\bar{y}$	$=27$ in	Cos $(\psi+\phi)$	$=-0{\cdot}207$
B	$=77$ in	μ	$=1{\cdot}0$.

In Case 3 the loads are:

Vertical
$$=1{\cdot}5\left[3R\left(1-\frac{\bar{y}}{2B}\tan\theta\right)+\frac{W\bar{y}}{2B}\right]$$
$$=1{\cdot}5\left[3\times612\left(1-\frac{27}{2\times77}\times0{\cdot}743\right)+\frac{2060\times27}{2\times77}\right]$$
$$=2940\text{ lb.}$$

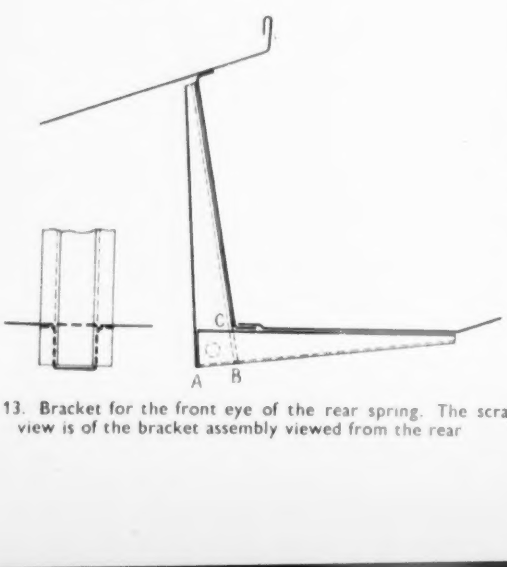


Fig. 13. Bracket for the front eye of the rear spring. The scrap view is of the bracket assembly viewed from the rear

Rearward
$$=4{\cdot}5R\tan\theta-0{\cdot}75W$$
$$=4{\cdot}5\times610\times0{\cdot}743-0{\cdot}75\times2060$$
$$=495\text{ lb}$$

Lateral $=0$.

In Case 4 the loads are:

Vertical
$$=1{\cdot}5\left[R\left(3+\frac{2\bar{y}}{T}\right)-\frac{\bar{y}}{2B}\left(\mu W\cos\psi+3R\tan\theta\right)\right]$$
$$=1{\cdot}5\left[610\left(3+2\times\frac{27}{43}\right)-\frac{27}{2\times77}\left(2060\times0{\cdot}208-3\times610\times0{\cdot}734\right)\right]$$
$$=3410\text{ lb}$$

Rearward
$$=4{\cdot}5\times610\times0{\cdot}743$$
$$=2040\text{ lb}$$

Lateral
$$=-\frac{1{\cdot}5\mu W\cos(\psi+\phi)}{\sin\phi}$$
$$=-\frac{1{\cdot}5\times2060\times-0{\cdot}207}{0{\cdot}407}$$
$$=+1570\text{ lb.}$$

The rear springs are each supported by two brackets, one on the heelboard and the other on a cross member formed by a step down in the boot floor. It will be assumed that the axle is positioned midway between the two eyes on each spring so that the vertical and side loads are shared equally by each end. Fore and aft loads can only be taken by the front attachment because the rear spring shackle is not capable of taking any. However, allowance must be made for the angle of the spring shackle at full bump, since this will have an effect on the fore and aft loads reacted at the front end of the spring.

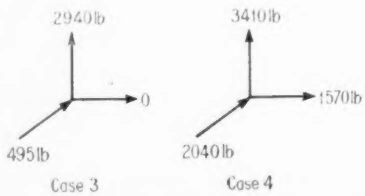


Fig. 12. Loads on the pin supporting the front end of the rear spring

Some of the vertical load will, of course, be reacted by the bump stops and shock absorbers, and further research may show that it is unnecessary to cater for the full load on the brackets carrying the spring eyes. However, it is considered advisable at present to design these brackets on the assumption that there is no such relief. The bump stops then should be made capable of carrying the full load less the spring rate multiplied by the deflection to full bump. The shock absorber loads are an entirely separate problem, and the manufacturers would, no doubt, be able to supply figures for the maximum loads that can be experienced.

It can be seen from Fig. 12 that the most severe loads are given by Case 4. Care is needed when deciding which case is the worst, because the direction of the resultant force is sometimes more important than its magnitude. For instance, were the resultant of the loads in the longitudinal vertical plane to be directed downwards, they would be much more critical because of the danger of the pins bursting out of their lugs: with the loads directed upwards they are supported by much more metal.

Each front spring eye bracket, Fig. 13, consists of two top hat sections, one with its flanges spot welded under the floor, parallel to the longitudinal axis of the vehicle, and the other with its base spot welded to the heelboard. The lower end of the bracket on the heelboard is extended into the rear end of the other one. The two brackets are spot welded together where they overlap.

At full bump, the rear spring shackle is at an angle of 27 deg from the vertical, so that the rearward directed component is:

$$\frac{3410}{2}\times\frac{1}{\tan(90-27)}$$
$$=870\text{ lb.}$$

This must be added to the drag load of 2040 lb, making a total of 2910 lb. The total horizontal load must be transferred to the body floor through $2910/275=11$ spot welds. In fact, because of the unreliability of welds and the likelihood of shock loading and severe fatigue at this point, it

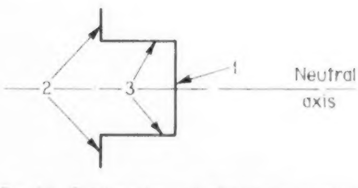


Fig. 14. Section through the upper portion of the rear spring bracket on the heelboard

would probably be safer to double this number. It is in making decisions on matters such as this that something more than mathematical ability is called for.

The load is a tensile one so far as the bracket is concerned, and to limit the stress to 16 ton/in² the sectional length of 18 s.w.g. sheet metal needed is:

$$\frac{2910}{16 \times 2240 \times 0.048} = 1.64 \text{ in.}$$

A top hat section 1·75 in wide × 1 in deep with ⅝ in flanges will be ample as it gives a cross sectional length of about 5 in. These dimensions are determined by the space requirement of the spring eye rather than by strength considerations. Where there is a possibility of the loads being reversed, it is advisable to check the compression strength. In this case, the applied stress is $(1.64/5.0) \times 16 = 5.25$ ton/in². The top hat section is obviously strong enough in compression to take this load.

Owing to the side load of 1570 lb, bending stresses will be superimposed. The load is distributed equally between the front and rear spring eyes, so that 785 lb is taken by the bracket under consideration. Initially, it is carried by the two lugs in bending to AB, Fig. 13, the horizontal base of the lower top hat section, and to the equivalent part, BC of the upper top hat section. The lugs are almost impossible to stress accurately, because membrane tension acts as a relief on the bending forces. However, a rough check can be made as follows.

When two lugs of a fork joint or a channel take side load it is usual to assume that 2/3 of the load is taken on each, so the load per lug is 524 lb. The length ABC is 3 in and the depth is $2 \times 0.048 = 0.096$ in, thus:

$$Z \text{ from } \frac{bd^2}{6} = \frac{3 \times 0.096^2}{6} = 0.046 \text{ in}^3.$$

With a mean moment arm of 2 in,

$$\frac{M}{Z} = \frac{524 \times 2}{0.046 \times 2240} = 10.2 \text{ ton/in}^2.$$

Some of this load will be carried by the underfloor bracket, but most will be taken on the bracket to the heelboard. The heelboard bracket is probably adequate by itself to cope with all the load, and if no undue weight penalty be imposed by making it strong enough to do so, this arrangement will be regarded as satisfactory.

The section is as shown in Fig. 14 and, because of its symmetry, the neutral axis may be determined by inspection. An approximate value for the moment of inertia is:

1. $\frac{bd^3}{12} = \frac{0.048 \times 1.75^3}{12} = 0.0214$
2. $Ad^2 = 2 \times 0.048 \times 0.625 \times 1.188^2 = 0.845$
3. $Ad^2 = 2 \times 0.048 \times 1 \times 0.875^2 = 0.735$

Total 1·6 in⁴

$$Z = \frac{1.6}{1.5} = 1.07 \text{ in}^3.$$

The maximum bending moment is at the junction of the bracket with the heelboard, and it is 1570 × 2 lb-in, so that the stress is:

$$\frac{1570 \times 2}{1.07 \times 2240} = 1.3 \text{ ton/in}^2.$$

This must be added to the stress due to the vertical loads.

The vertical load is reacted by the spot welds joining the heelboard and the top hat section. Because of the slope of 20 deg of the heelboard, this reaction is 3410/2 cos 20 deg = 1815 lb, and the number of spot welds required is 1815/275 = 7. Here again, it will be advisable to double this number. Moreover, at the bottom, where the fairly heavy and concentrated shear due to side load must be reacted, about ¾ in of gas weld will be necessary on each side of the bracket to join it to the toe board so as to assist the spot welds at this point. Next, the top hat section must be checked for compression strength as a strut with varying end load. Since the section is similar to that of the horizontal portion of the bracket, the stress will be approximately $\frac{1815}{2910} \times 5.25 = 3.27$ ton/in². It will be noticed that the slope of the heelboard bracket introduces a horizontal component which will be reacted by the underfloor bracket. However, this is a small component that is in any case a relief on the rearward drag load. Moreover, the bracket is relatively lightly stressed, so this component was not taken into consideration when checking the underfloor bracket.

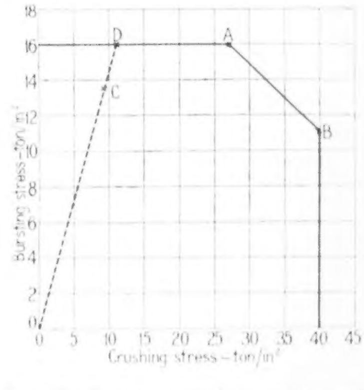


Fig. 15. Design curve for crushing and bursting stresses

Next, the pin and the holes in which it is carried in the lug must be checked. It is a generally accepted rule when checking pins for strength in shear that if the width between the planes of shear, divided by the pin diameter, is less than 2, then the formula to be employed is $f = \frac{P}{1.75A}$. If, on the other hand, the width divided by diameter is greater than 2, the formula is $f = \frac{P}{2A}$, where f is the shear stress, P is the shear load and A is the cross sectional area of the pin. In this case, $P = \sqrt{\left(\frac{3010}{2}\right)^2 + 2040^2} = 2540$ lb.

Assuming that the shear stress in the pin is to be limited to 10 ton/in², the diameter of the pin must be given by

$$\sqrt{\frac{4 \times 2540}{10 \times 2240 \times \pi \times 2}} = 0.269 \text{ in.}$$

This is obviously too small a diameter for a shackle pin, so bending of the pin must be the criterion. Rubber bushes are to be employed so that it probably will not be far from correct to assume a uniform distribution of loading over the 1¾ in length of the pin between its bearing lugs. The bending moment M is given by

$$\frac{Wl}{8} = \frac{2540 \times 1.75}{8} = 554 \text{ lb-in}$$

and the stress by:

$$\frac{M}{Z} = \frac{554}{0.024 \times 2240} = 10.3 \text{ ton/in}^2,$$

where the figure 0·024 is the modulus of section Z of a ⅝ in diameter pin. Although this is a fairly low stress, it will leave a safety margin to allow for errors due to the assumption of a uniform distribution of load.

The holes for the pin in the lugs must next be checked. There are two possible causes of failure: one is crushing and the other is bursting. That is to say, either the material around the hole may crush, or the pin may burst out of the hole. Each pin is supported by two lugs, both of which are made up from two thicknesses of 18 s.w.g. (0·048 in) sheet steel.

$$\text{Crushing stress} = \frac{2540}{2 \times 2 \times 0.048 \times 0.625 \times 2240} = 9.4 \text{ ton/in}^2.$$

There is ½ in of metal around the hole so that the bursting stress is given by:

$$\frac{P}{1.75\,(R - r) \times t} = \frac{2540}{1.75 \times 0.5 \times 2 \times 0.048 \times 2240} = 13.5 \text{ ton/in}^2$$

where P = the bursting load
R = the radius of the outer periphery of the lug
r = the radius of the hole
t = the thickness of the lug.

Permissible crushing and bursting stresses are interdependent, and figures such as the following for T4 tube may be obtained from tests. Maximum safe bursting strength is 16 ton/in^2 and it is obtained when the crushing stress is 27 ton/in^2 or less, while the maximum safe crushing stress is 40 ton/in^2, and it is obtained when the bursting stress is 11 ton/in^2 or less. From this data a curve may be drawn, Fig. 15, the points A and B representing respectively the first and second pairs of values just quoted. Then the actual crushing and bursting stresses are plotted at point C.

In order to demonstrate the use of the curves, the figures 9·4 ton/in^2 and 13·5 ton/in^2 are plotted on the T4 curve, although this, of course, would not be the material used in this particular case. A line is then drawn through the origin and point C, and extended until it cuts the curve at D. If point D lies between A and B, its co-ordinates represent the permissible crushing and bursting stresses. In this example, it is on the horizontal part of the curve, thus indicating that bursting is critical and that the maximum permissible bursting stress is 16 ton/in^2. Another check must be made on the bearing stress in any special bushes employed. The allowable stresses may be obtained from the suppliers of the bushes or of the material from which they are made. Alternatively, it may be determined from mechanical tests.

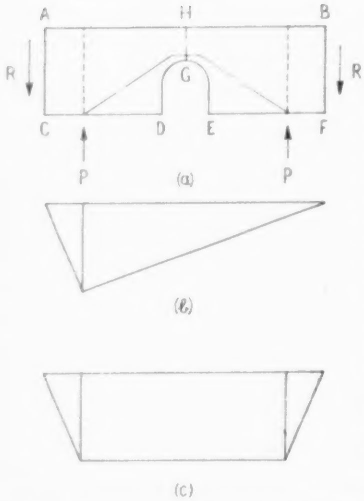


Fig. 16a. Diagrammatic illustration of the heelboard arrangement
b. Bending moment diagram when the load is applied to the bracket on one side only
c. Bending moment diagram when the load is applied to both brackets

The final check on the front anchorage point of the rear spring is to ensure that the heelboard is strong enough to carry the bending loads. An unfortunate feature of this component is the cut-out for the propeller shaft tunnel. The system of loads is depicted in Fig. 16a, in which it can be seen that the applied vertical loads P are balanced by the reactions R at the quarter panel. The type of bending moment diagram for Case 4 is given in Fig. 16b, and that for Case 3 in Fig. 16c. Section GH is obviously critical so far as bending is concerned, and Case 3 will be the design case. The loads P will be 2940 lb and the reactions R must be equal and opposite. Since the distance between P and R is in each case 5 in, the maximum bending moment is $5 \times 2940/2 = 7350$ lb-in.

For the purpose of designing the heelboard, it will be assumed that no relief is offered by the weight of the passengers in the rear seat, because equally severe loads could probably be experienced with an overloaded luggage boot and the rear seats empty. The bending moments will induce a tensile load in the top flange and a compressive load in the bottom flange of the heelboard. However, the gap for the propeller shaft tunnel makes it impossible to maintain the continuity of the bottom flange. As a result, the compressive end load must be bypassed over the top of the gap.

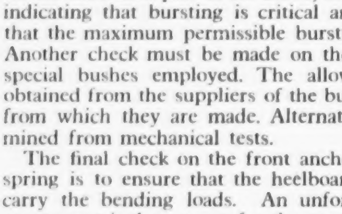


Fig. 17. Section GH, Fig. 16a

There are two ways of doing this. One is to spot weld horizontally on to the front face of the heelboard, a reinforcing channel section of depth GH with its upper flange immediately below the seat pan, and its lower one above the tunnel. This channel should extend the full width of the heelboard, and could be hidden by the seat upholstery. Its function would be to relieve the flanges CD and EF of all the bending load. The other method, although somewhat more complex, would result in a lighter structure. It is to carry the end loads from the lower flange where it crosses each suspension bracket by means of a channel or angle section spot welded to the heelboard and passing over the cut-out. The path this member must take is shown chain dotted in Fig. 16a. If the top of the propeller shaft tunnel were flattened and, together with the flange of the heelboard to which it is attached, spot welded to the member, it would afford considerable reinforcement at the weak section.

This method of carrying the loads was stated to be more complex than the first because of the need for a gas welded attachment at each lower end of the member, and for two kink struts at the top where the sloping portion on each side joins the horizontal part over the tunnel. It might be simplified to some extent either by pressing the member and kink struts in one piece or by using swages in the heelboard for kink struts. Neither method is very satisfactory inasmuch as neither provides support for the lower flanges. Moreover, there is always a chance of fatigue failure originating at the ends of swages. However, in this case, support for the flanges might be obtained by welding the corners to the tunnel. Incidentally, the term swage is applied to a channel section pressed into a sheet of metal. Such a device is commonly used in press work, particularly in large flat sheets with little or no curvature, to take up loose metal which might otherwise give rise to drumming.

Although the overall depth of the section GH is 3 in, Fig. 17, its depth between the neutral axes of the flanges is only 2¾ in, and the bending moment is 7350 lb-in. This moment would induce in the flanges an end load of approximately $7352/2\frac{3}{4} = 2680$ lb. Assuming that the compressive stress must be limited to 12 ton/in^2, the area of metal needed to carry this load is 0·1 in^2. Therefore 2·8 in of 20 s.w.g. metal will be required in the top flange. This requirement is in fact covered by the following 16 t widths of metal: the turned-up flange of the seat pan, one on each side of the spot welded joint between the seat pan and the heelboard, the flange of the heelboard, and the portion of the heelboard stabilized by its flange. The total is $5 \times 0{\cdot}575 = 2{\cdot}88$ in.

On the tension flange the allowable stress is 16 ton/in^2, and the width of 20 s.w.g. sheet needed is $(12/16) \times 2{\cdot}88 = 2{\cdot}16$ in, which may be provided by four 16 t widths of sheet. This area of metal requirement is also met even without taking the long tension member into consideration. The four widths are as follows: two on the propeller shaft tunnel, one on the heelboard flange spot welded to the tunnel, one on that part of the heelboard which is stabilized by the flange. Incidentally, it is wisest to design for a reversal of load, so that the tension member also should be

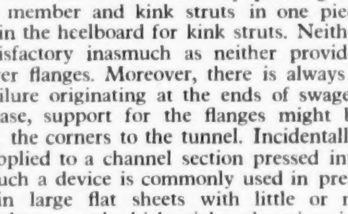


Fig. 18. Shear distribution on a beam

checked as a strut by estimating its L/K ratio and crippling stress.

In Case 3, the shear on section GH is zero. In Case 4, with moments taken about one end, it is $(5/47{\cdot}5) \times (3410/2) = 180$ lb where 47·5 in equals the total width of the heelboard and 5 in is the distance between the centre of a spring bracket and the adjacent end. The unreinforced cross section at GH is shown in Fig. 17. In this type of structure, where a beam has a thin sheet metal web that is very deep relative to its flanges, it is usual to assume that the shear is all uniformly distributed over the whole depth of the web. Fig. 18 represents the type of shear distribution diagram calculated from the $SA\bar{y}/IB$ formula for such a beam. From this diagram it can be seen that the assumption of uniform distribution of shear over the web is justified. In fact, the error is never more than 5 per cent, and is often considerably less. On this basis, the shear stress is $180/(3 \times 0{\cdot}036) = 1665$ lb/in². This stress will be satisfactory. The subject of allowable shear stresses in panels will be dealt with in greater detail in a later article, but for rectangular panels it is safe to design for a limiting stress given by the formula:

$$f_s = \frac{kE}{\left(1-\nu^2\right)\left(\frac{t}{b}\right)^2}$$

where f_s = the limiting shear stress
E = Young's modulus
ν = Poisson's ratio
t = the thickness of the sheet
b = the length of the shortest side of the panel
a = the length of the longest side of the panel
k = a constant from the table below

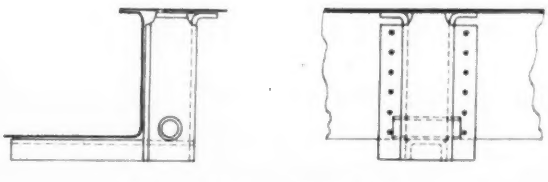


Fig. 19. Rear spring shackle bracket

Next the rear shackle attachment bracket and cross member must be checked. Shortage of space prohibits a full description of this process, but there are one or two outstanding features, of this part of the design, worthy of special note. The cross member is only 3½ in deep so that the stresses will be fairly heavy. Its web should be stiffened with swages positioned vertically about $\frac{3}{4} \times 3\frac{1}{2} = 2\frac{5}{8}$ in apart. In general, swages employed to stiffen a beam should be spaced three quarters of the depth of the beam apart along its length. However, this is not a rigid rule and it may be varied to suit different conditions. It is important that the ends of the swages should be more than 16 t from the flanges, otherwise the capacity of the flanges for carrying end loads will be reduced, and there will be a danger of fatigue cracks originating at the ends of the swages.

With regard to the brackets, there is one important consideration. Because the line of application of the load is offset from the vertical web of the cross member, and also because of the angularity of the shackle, there is need for horizontal shear attachments at the top and the bottom of the bracket. In other words, the top of the bracket must be flanged and spot welded to the boot floor, and a rearward extension piece must be employed to attach the lower end of the bracket to the rear part of the boot floor, Fig. 19.

If the rear of the extension piece does not come near the vertical rear wall of the boot, a swage should be pressed in the floor extending over its full width and passing over the back end of the extension pieces at each side. This swage completes the structure necessary to take the loads from the extension pieces.

To be continued

a/b	1·0	1·2	1·4	1·5	1·6	1·8	2·0	2·5	3·0	∞
k	7·75	6·58	6·0	5·84	5·76	5·59	5·43	5·18	5·02	4·4

HOT SPRAYING

An Improved Finishing Technique

AN ingenious alternative to the use of volatile thinners for reducing the viscosity of finishing enamels is a device which heats the paint on its way to the spray gun. The new process, which is of American origin, should be of particular interest to the automobile industry since it appears to solve a number of temperature and humidity problems and needs only one coat to give a perfect surface. Furthermore, it saves solvent costs, greatly reduces blushing and practically eliminates runs.

The "Control-Temp"

Built into the spray line the "Control-Temp," as the new device is called, is a small cylinder about eight inches long and weighing a pound and a half. Inside the cylinder is a helical tube through which steam at five or ten pounds pressure or water at 200 deg F is circulated. The finishing material passes round the outside of this tube on its way to the gun. The cylinder is completely insulated so that it can be held in the operator's left hand while he works the gun with his right. Alternatively, it can be hung from a waist belt or even dragged along the floor, for it is equipped with heavy-duty rubber bumpers to withstand rough usage. This appliance is manufactured by the Arco Company, Cleveland, Ohio, U.S.A.

The "Control-Temp" uses a standard spray gun with room temperature air pressure. One great advantage is that no special circulating equipment is needed and only the finishing enamel actually used is heated. Although only a small quantity of finishing material is in the heating unit at any particular moment, heating action is so rapid that it is quite easy to maintain an output of one and a half pints per minute. There is an economy in operator's time since there is no warm-up period needed.

Cleaning is normally done by flushing the heater with solvent. At intervals, the unit will benefit from an alkaline cleaning of the outer shell through which the finishing medium has passed. Where there is no heat source available, a small steam generator or a 35 gallon hot water tank with circulating pump will normally be adequate. It would appear that this equipment has great potentialities. Claims are made that when it is used the quality of the work is of a more uniform and better quality than that usually produced with the more commonly used equipment, while at the same time it can also lead to appreciable economies.

PLASTICS BODIES

Some Recent Developments in the United States of America

IT would appear that interest in reinforced plastics for automobile bodies has become widespread in the U.S.A. In an article published by *Modern Plastics* in their February 1953 issue, there are illustrations depicting nine different makes of plastics bodied cars, and some details are given concerning a number of others. Nearly all the cars described are sports or convertible models for relatively small quantity production.

However, some of the larger manufacturers are also experimenting with this material. Howard Darrin, who has styled some of the Kaiser models, is now designing several prototypes from which the production model of the Kaiser sports car will be selected. One of these incorporates features such as a convertible top which retracts into the boot compartment, and doors which slide instead of swinging out on hinges. Sliding doors may be more practicable with plastics than with the more conventional steel construction. This is certainly a feature which has advantages to a motorist parking either in busy streets or in a small garage.

Kaiser-Fraser state that this particular vehicle is not the production version of the new K-F sports model which will be officially introduced at the Chicago Auto Show in March. Production plans for this new sports car and details of the moulding procedure to be followed have not yet been announced. It is also rumoured that Chevrolet may be producing a plastics bodied Corvette.

Another car that may be shown at the same time is the new Buick Wildcat. Ivan L. Wiles, general manager of the Buick Motor Division and vice president of the General Motors Corporation, has announced that the Wildcat was designed to test the possibility of using glass fibre plastics for bodies. The use of this material could lead to greater flexibility in styling since the tooling costs for low volume production would probably be considerably less than with more conventional materials. In fact, some authorities state that these costs are only 10 per cent of those of press tools for steel bodies. Unfortunately, the material cost of glass reinforced laminated plastics in America is as much as 10 times that of sheet steel, and only when manufacture is on a small scale can advantage be taken of the low tooling costs that may make the use of plastics an economic proposition.

Improvements are certainly being made in production processes. By using a single blanket of uncured material and a suitable press, panels can be made in as little as three minutes. This is still appreciably longer than the time required to form a steel panel. On the other hand, it is not necessary to perform extra operations to reinforce plastics panels and give them adequate stiffness, whereas the time for additional fabricating work associated with the stiffening of a bonnet lid, for instance, must also be taken into consideration.

The Muntz Car Co. Inc., of Evanston, Illinois, is planning to place a reinforced plastics car body in production early in 1953. This will be used for the Muntz Jet sports model. Contrary to the usual practice with bodies made from plastics material, this will have individual moulded wings, canopy, rear deck and doors produced in matched mould dies. These components will be assembled to make the finished car. It is hoped that this arrangement will not only further facilitate the already simple process for repair of accidental damage, but that it will also reduce the labour costs involved in hand production of one piece bodies.

A large number of small manufacturers in America have also produced plastics bodies. Among these is the Glasspar Co. of Santa Ana, California, who make the Boxer. Vale Wright of Berkeley, California, plans to turn out one complete body per day, suitable for mounting on the M.G. chassis. Viking-Craft, of Anaheim, California, are now producing two glass fibre reinforced plastics models, one to fit the chassis of the 1947-1951 Crosley chassis and the other is the Super Skorpion body designed for the Super Sports Crosley, the Renault, Morris and other small chassis. The weight of this unit is claimed to be less than 150 lb. Another made by the same manufacturer is known as the Cheetah, but it has yet to be shown to the public. It is to fit a modified Ford chassis. A coupé type body for the M.G. chassis is being made by Atlas Fiber-Glass Inc., of Alhambra, California. In *Modern Plastics*, it is stated that this model is available with either one or two doors. An unusual structural feature is that the body, which weighs less than 200 lb, is reinforced with [illegible] in steel tubing.

In Detroit, Gino and Cesare Testaguzza have made a reinforced plastics sports car body for mounting on a Chevrolet chassis. This body is called La Seatta. It is made in one piece, including the front bumper, and weighs 157 lb. The Ray Greene Co., of Toledo, estimate that with Plaskon polyester resin and glass fibre, they could step up production to four bodies a day within thirty days, and to twenty per day within sixty days. The model they produce is mounted on a Henry J. chassis.

The Rockefeller Sports Car Corporation, of Rockville Centre, New York, is making a four seater sports car body based on a Ford V-8 chassis. Lunn Laminates, also of New York, do the moulding. This body is made in four parts: the front and rear ends, and two doors. The finished panels are approximately $\frac{3}{16}$ in thick.

Another use for reinforced plastics has been found in styling departments. Clay mock-ups of new models are not very robust, and it is difficult to simulate satisfactorily the finish of an actual car. However, plastics shells made from moulds produced from the clay models will last almost indefinitely, and they can be metal-sprayed or painted to give the appropriate finish. In fact, it may well be that the rumours current in America concerning the production by large manufacturers of plastics bodied cars arise from the fact that these firms are now displaying some plastics prototypes instead of steel ones at motor shows. This would be a relatively inexpensive means of testing public reaction to the style and general conception of a model before investing capital in press tools. In that way a lot of the risk associated with the introduction of new models would be circumvented.

DE-RATING GASKETS

THE Alexander Engineering Co. Ltd., of Haddenham, Bucks., are producing thick cylinder head gaskets for lowering the compression ratios of engines to adapt them for low octane fuels, or for conversion from petrol to vaporizing oil fuels. These gaskets are made from a thick sheet aluminium washer with a Plexeal aluminium foil lamination on each side. The laminations are bonded to the sheet by means of a thermo-setting resin adhesive. This arrangement provides the flexibility necessary for the gasket to adapt itself to joints that are not absolutely flat. However, the manufacturers emphasize the fact that this gasket is not intended for use in badly distorted joints. Aluminium foil ferrules are fitted round the cut-outs for the cylinder bores, and other holes guard against accidental peeling and tearing of the laminations during handling.

It is claimed that these gaskets are superior to the conventional copper and asbestos types because of their better thermal conductivity. This allows heat to be conducted from the head to the cylinder walls which otherwise are sometimes over-cooled. It also helps to eliminate hot-spots which cause pinking and running-on. When thick gaskets are fitted it is, of course necessary to make some adjustment to the push rod length. (1038)

SMALL CAR PRODUCTION

The Manufacture of Engines and Transmissions for the Austin 'A30'

Part I

SEVERAL developments of outstanding interest are included in the equipment for producing the power units and transmissions for the Austin 'A30,' which is inevitably more generally known as the Austin '7.' The engine for this vehicle is a four-cylinder unit of 58 mm bore and 76 mm stroke, with a three-bearing, partly balanced crankshaft and push rod operated overhead valves. Because of its outstanding performance for its capacity, this engine is also to be used as the power unit for the Morris Minor. The decision that this engine should be common to both the '7' and the 'Minor' has entailed arrangements for doubling the output for which the machine lines were originally planned. These notes are based on the original tooling, but they contain references to certain changes that are to be made to obtain the increased output.

So far as general machining is concerned, the outstanding features of the tooling are the use made of multi-station in-line transfer machines and of unit machine heads designed and manufactured by the Austin Motor Co. Ltd. As is well known, multi-station in-line transfer machines are in use in several automobile factories, but in the Austin works, transfer machining is applied to a much wider range of components than in any other British automobile factory.

Unit machine heads

Machines of the Company's own design are not unknown in other organizations, but they are generally for specific applications for which standard machines are not suitable. Austin unit machine heads, on the other hand, have been developed for use on a wide range of machining operations. They are now in use on single station and multi-station machines for several of the engine and transmission components for milling, drilling, reaming and tapping operations.

The manufacture of the unit heads has been organized on a quantity production basis, and the designs are such as to give maximum efficiency at minimum cost. As the products to be machined are stabilized, once a head is put into use it will be used for the same operation for a considerable period. There is, therefore, no need to make provision for quick changes over a wide range of speeds and feeds.

A relatively small range of unit heads can be adapted for a wide variety of work. In basic design all the heads are generally similar. Each is so designed that it is suitable for right- or left-hand mounting in a horizontal, vertical or angular plane, and incorporates mechanism for an automatic cycle of fast approach, feed and rapid return. Essentially, a unit comprises a spindle head mounted on a saddle that moves on a slide which forms the base. A two-start feed screw supplies the feed movement. The spindle head embodies its own drive motor, and the normal feed drive is taken from the spindle through a reduction gear. Rapid traverse forward and return movements are controlled by a separate motor in a fast-feed unit attached to the extreme end of the base slide. Micro-switches on the base, operated by trip dogs on the saddle, control the automatic cycle. Through contactor relays, these switches can be arranged to start, stop and reverse the spindle and fast-feed motors in any desired sequence.

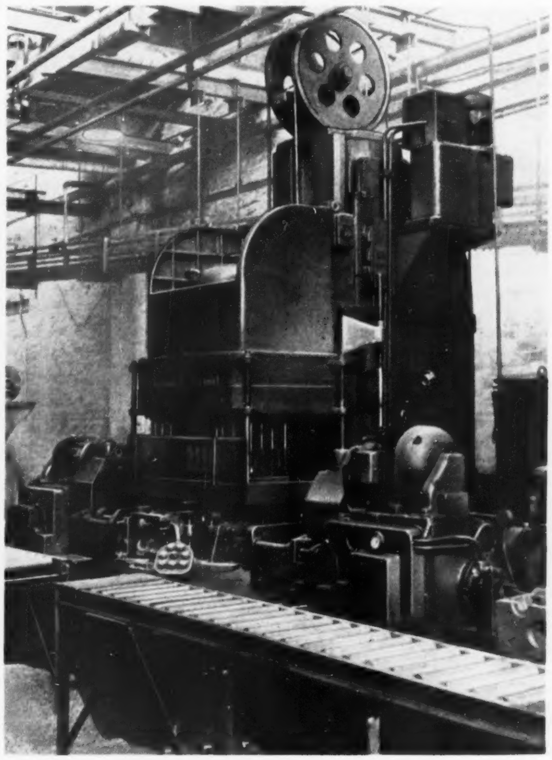


Fig. 1. Archdale three-way drilling and boring machine with four-station rotary indexing table

Substantially, the drive arrangement is the same for every size of head. The drive to the spindle from the spindle drive motor is taken through a series of pick-off gears on splined shafts. Pick-off gears are used for both speed and feed changes, but the changes are not so easily effected as on conventional machines. There is, of course, no need for frequent changes of feeds and speeds. Once a head has been allocated to a specific job, it is geared to give the optimum machining conditions, and normally it will be used for the same job until change in design calls for a re-tooling. Relatively complex and expensive means for rapid changes over a wide range of feeds and speeds are unnecessary and would be uneconomic.

The micro-switches for controlling the automatic cycle are mounted on both sides of the base slide. Four Burgess MK 4BR micro-switches in three lanes are mounted in the standard switch box. This gives a normal cycle of fast approach, feed and rapid return. The switches and dogs controlling the extreme positions of the head are duplicated on each side and are connected in series as a safety measure.

More complex automatic cycles can be obtained by using more dogs or by using the switch boxes on both sides of the unit. On a head that is to be used for tapping operations, one of the switches is arranged to reverse the spindle motor. This gives the following cycle: rapid approach, feed, reverse spindle feed, rapid return and stop.

B.T.H. rotor-stator units have been adopted as the driving motors. Different speeds and powers can be obtained by changing the rotor and stator in the housing. So that units with different strokes can be assembled to suit various applications, base slides and lead screws are produced in a series of standard lengths for each size of head. A driving dog for driving a multi-spindle head or other attachment is incorporated in the spindle nose. The attachment is bolted to the face of the unit spindle head, which is jig drilled with bolt holes and dowel holes in standardized positions to allow any appropriate attachment to be mounted. There are also

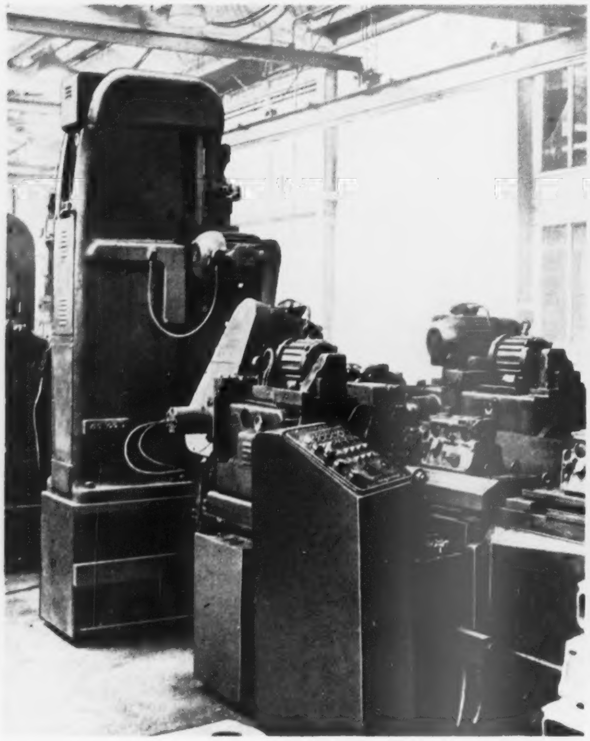


Fig. 2. Control panel and loading station for a 12-station in-line transfer machine for the Austin "A30" cylinder block

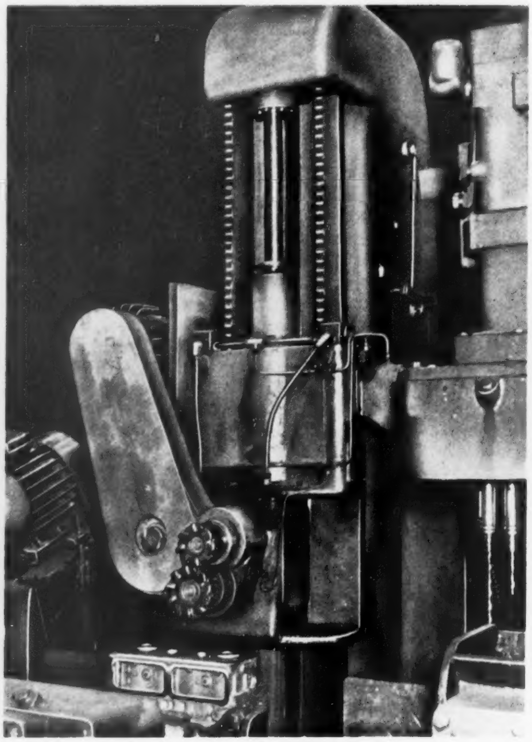


Fig. 3. Milling head for machining the tappet cover face and the petrol pump facing on the machine shown in Fig. 2

alternative positions for mounting the unit head on the saddle to allow for the length of a multi-spindle head or other attachment. All unit spindles are bored to a standard taper so that single point tools can be used where required.

Cylinder block machining

On the first machine in the cylinder block line the joint and sump faces are milled to leave 0·025 in of stock to be removed at subsequent operations. Originally, the blocks were machined singly on a duplex horizontal miller tooled to machine both faces simultaneously. To meet the increased output requirements, an Asquith vertical rotary milling machine with 10 fixtures has been installed. Five of the fixtures are designed to take the casting as received with the joint face up; the other five give location from the machined joint face for milling the sump face. Roughing and semi-finishing cuts are taken on both faces at a surface cutting speed of 250 ft per minute.

From the first operation, the block is transferred to an Archdale four-station rotary indexing transfer machine, see Fig. 1, with three working stations and one loading and unloading station. This machine has a vertical head that operates at all three working stations and two horizontal heads, each of which operates at two working stations. The component is loaded with the sump face up.

At the first working station, the vertical head drills two location holes in the sump face, six holes for the main bearing cap bolts, an oil suction hole, two sump screw holes and part drills the dipstick hole. At the same time the left-hand horizontal head operates to drill the dynamo mounting hole and part-drill the oil release valve hole.

The table is indexed, and at the second working station the two location holes in the sump face are reamed from the vertical head. These holes act as register points for subsequent operations. The vertical head is also used at this station for counterboring the six bearing cap holes and the oil suction hole, and for drilling the dipstick hole through. Both horizontal heads are in use at this station, each drilling the main oil gallery hole to a depth of 3½ in from the end.

At the final station on this machine the vertical head and the right-hand horizontal head are in action. From the vertical head the bearing cap holes are reamed, 12 screw holes are drilled in the sump face and the distributor shaft oil hole is drilled. The right-hand horizontal head is used for counterboring three off-side welch plug holes and for drilling the oil release valve hole to depth.

At the next operation eight tappet guide holes are drilled. The output originally planned could be maintained from one machine, an Archdale four-spindle vertical drill, but to meet the increased output, an Austin four-spindle unit head machine will also be employed. Similar methods will be used on both machines, which have identical work fixtures. The centre distance between the inlet and exhaust tappet holes for any one cylinder is too close to allow them to be drilled simultaneously, but provision is made for all eight holes to be drilled at one loading.

The work is loaded on to an in-line three-station indexing fixture. At the loading position the work is central under the machine head. When the casting is located, air clamps are applied. The table, and with it the fixture and the work, is pneumatically indexed a short distance to the right to bring the casting into the desired position for drilling four of the holes. After these holes are drilled and the head is retracted, the table is again indexed to bring the work to the third position, which is left of centre for drilling the other four holes. Finally, the table is indexed to the original position and the block is unloaded.

Before the next operation the casting is turned over on to the sump face. It is then loaded into a three-way drilling and boring machine in which the cylinder bores are rough machined from a four-spindle vertical boring head. At the same time, the rear end welch plug hole is counterbored from a horizontal head at the rear and two tappet cover holes are drilled from a horizontal head at the right. The main oil gallery hole is then drilled through from each end on an Austin unit head machine.

An inspection station is located immediately subsequent to this operation. Special receiver gauges have been designed to facilitate percentage inspection at appropriate stages in the machining sequence. It is, of course, not only necessary to ensure that every machined element is dimensionally correct but also to ensure that the spatial relationships between the different elements are maintained within the specified tolerances. The receiver gauges allow the necessary checking to be carried out quickly and accurately. At this stage, the gauge is designed to check the top and bottom face distance, the location holes, bearing cap bolt holes, counterbores, oil suction hole depth, the rough machined cylinder bores, the rear end welch plug hole and the tappet cover stud holes. Three welch plug holes are then fitted in the off-side and the casting is given a water test.

From the water test the casting is transferred to the first of a number of in-line automatic transfer machines. Before the operations carried out on this machine are described, some general remarks may be made about the transfer machining methods generally in this organization. The transfer machines themselves are in some instances completely designed and manufactured by James Archdale and Co. Ltd.; in others, Austin unit machine heads are used in conjunction with bases and transfer mechanisms of Archdale design and make.

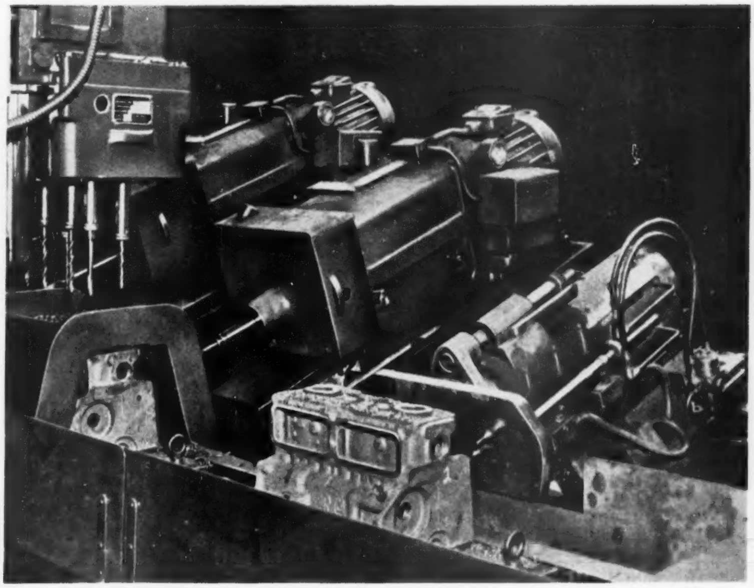


Fig. 4. Blow-out and gauging unit on the transfer machine

In one respect, the practice adopted by the Austin Motor Co. Ltd. for in-line automatic transfer machining differs from that employed generally by other automobile manufacturers in this country. This is, that without exception the component is mounted on a platen, or carrier, plate for its journey through the machine; this is the exception rather than the rule in other organizations. There are certain disadvantages in the use of a platen plate. One is that loading and unloading take more time than is required when the component is loaded direct into the machine and is automatically released at the end of the machine. A second is that the plate must be returned to the loading station. To set against these disadvantages is the fact that it is easier to ensure accurate register of the plate at each successive station, and very much easier to clamp it securely than it is to register and clamp a casting of complex shape. In addition, the wear on register points such as the location holes in the sump face of the cylinder block is greatly reduced, and there is therefore much less danger that small but significant inaccuracies may occur. After all the pros and cons had been considered, it was decided to use platen plates on all in-line automatic transfer machines.

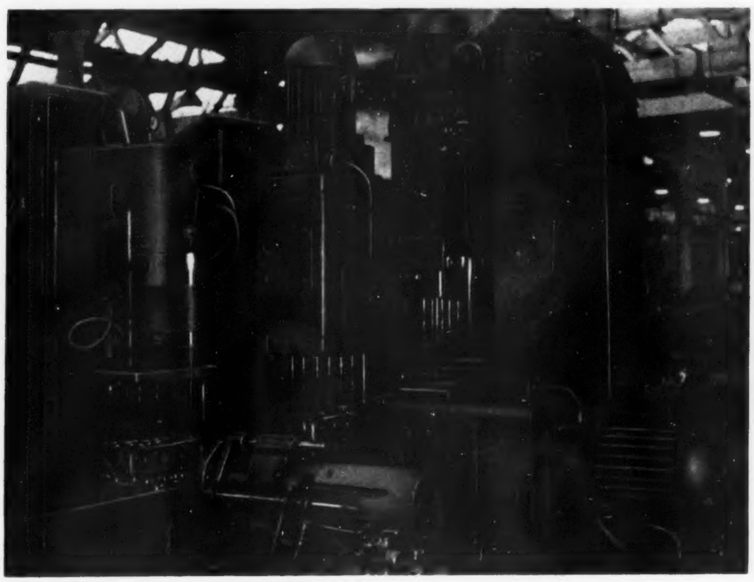


Fig. 5. Blow-out and gauging units for the cylinder head stud holes and four holes in the side of the casting

Platen return is at present effected on an enclosed overhead circuit with King electric hoists for raising the platen and conveying it from the unloading to the loading station. This is an effective and economical method, but it has certain drawbacks, and it is probable that alternative arrangements will be adopted. Swarf disposal is another problem that is inseparable from the use of multi-station in-line automatic transfer machines. In this factory, the method adopted is to have an endless slot conveyor running under the machine and arranged to deposit the swarf in a receptacle at the end of the machine.

The first transfer machine in the line as laid out at present is an Archdale 12-station machine, but to cope with the increased production a 23-station machine will be installed. These notes deal only with the existing machine. For passage through this machine, the block is placed sump face down on the platen fixture with the rear end leading. Location is taken from the reamed holes in the sump face and the casting is clamped through the first and fourth cylinder bores. A pilot bush for the distributor shaft hole is incorporated in the fixture.

From the loading station, Fig. 2, the casting is indexed to an idle station and thence to the third station where the left-hand unit is a travelling head milling machine with a 3 in diameter face milling cutter for machining the distributor boss. The right-hand unit at this station, Fig. 3,

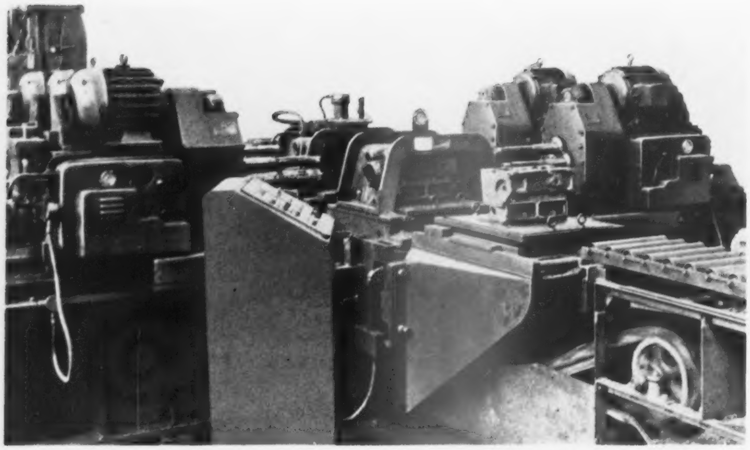


Fig. 6. Five-station transfer machine with boring spindles at each working station

is also a travelling head milling machine, but with two spindles, one for milling the tappet cover face and the other for milling the petrol pump facing.

At the fourth station, the left-hand unit comprises a rise and fall milling head mounted on a standard Archdale column for a vertical drilling machine. It carries two spindles with 2½ in diameter face milling cutters for machining the dynamo mounting bosses and the oil filter boss. The right-hand unit at this station is a four-spindle drill head. It drills two holes in the petrol pump face, a hole for the breather pipe clip and a water drain hole.

There are two drilling units at the fifth station. That at the left is a vertical 11-spindle unit tooled for drilling nine cylinder head stud holes, an oil hole and an oil pump priming hole. At the right there is a single spindle horizontal unit for drilling the water drain hole through. The sixth station has an angular two-spindle drill unit at the left for drilling the distributor set pin hole and a hole through into the main oil gallery. The right-hand unit at this station is a two-spindle drill unit tooled for spot-facing the water drain boss and the breather pipe clip boss.

At the seventh station there is a left-hand angular two-spindle unit tooled for drilling the large distributor hole and for spot-facing the oil gauge union boss. The right-hand unit at this station is a vertical 11-spindle drill head used for drilling two push rod holes, six water course holes, two rivet holes for the engine number plate and for pin facing the oil priming hole. At the eighth station the distributor shaft hole is drilled from a left-hand angular unit, while from a seven-spindle vertical right-hand unit two push rod holes are drilled, other two push rod holes are drilled and reamed and three water course holes are drilled.

Many of the holes that have been drilled up to this stage have to be tapped, and later stations on this transfer machine are tooled for tapping. To eliminate the danger of tap breakage, it is essential to ensure that blind holes are drilled to the correct depth. It is also essential that the holes be cleared of swarf, both to ensure that a build-up of swarf does not in effect reduce the depth of a blind hole, and to prevent the production of incorrect threads. For this reason, the ninth and tenth stations are occupied by special blow-out and checking units, shown in Figs. 4 and 5. Each unit has a series of nozzles so arranged that they enter the appropriate holes when the head advances, and compressed air passing through the nozzles blows out the swarf. The nozzles also act as depth gauging units. At the ninth station there is only one horizontal angular unit arranged for blowing the swarf out of the distributor set pin hole and for gauging the depths of the set pin hole and the oil gauge union hole. A vertical unit at the left-hand side of the tenth

Fig. 7. 12-station in-line transfer machine with drilling, reaming, milling and tapping heads

station is used for blowing out and checking the depth of the cylinder head stud holes and the oil pump priming hole, while the right-hand horizontal unit is used to blow-out and check the depth of the tappet cover stud hole, the petrol pump stud holes and the breather pipe clip hole.

There are two multi-spindle tapping units at the eleventh station, one vertical for tapping nine cylinder head stud holes and the oil pump priming hole, while a horizontal unit taps tappet cover stud holes, petrol pump stud holes, the water drain hole and the breather pipe clip stud hole. At the final working station on this machine a two-spindle angular unit is used for tapping the distributor set pin hole and the oil gauge union hole. There is also a vertical multi-drill head at this station. It is tooled for re-drilling an oil hole through to the front cam bearing, two push rod holes and five water course holes. To deal with the larger output that is now planned, a 23-station in-line automatic transfer machine will be used. It is designed to perform exactly the same operations as the existing machine but it will give double the output.

The casting is then transferred to a second in-line transfer machine, see Fig. 6. For passage through this machine, which has five stations, the block is mounted on the platen plate with its longitudinal axis at right angles to the longitudinal axis of the machine and with the sump face up. Location is taken in the work fixture from the sump face and the location holes. The machine has boring units at each of the working stations.

At the first station the left-hand unit carries two spindles, one tooled for boring and facing the oil pump recess and rough boring and facing another diameter, while the second rough bores the rear half crankshaft bearing. The right-hand unit at this station has three spindles. One is tooled for rough and finish machining the outer water pump bore for diameter and depth, for rough machining the inner water pump bore, and for chamfering the outer water pump bore. The second rough bores the front cam bearing, and the third rough bores the front half crankshaft bearing.

From the left-hand unit at the third station the rear camshaft bearing and the centre half crankshaft bearing are rough bored, while from the right-hand unit the centre camshaft bearing is rough bored. It should be noted that the camshaft bearing diameters vary, the front being the largest and the rear the smallest. This is advantageous when semi-finish and finish in-line boring are carried out at a later stage. Only one boring unit is mounted at the final working station on this machine. It is used for finish facing the oil pump recess. Between this machine and the next, there is a receiver gauge placed conveniently in relation to the roller conveyor between the machines. It is designed for checking all the operations carried out on the two preceding in-line transfer machines.

At this stage the casting is passed to another Archdale in-line automatic transfer machine, shown in Fig. 7. This one has 12 stations including the loading and unloading stations. The component is loaded on the platen with the sump face up and with its longitudinal axis in line with the longitudinal axis of the machine. At the first working station, an angular unit at the left-hand side is used to spot-face and chamfer the distributor pilot shaft boss while from the right-hand side a rise-and-fall milling head, see Fig. 8, mills the inner sides of the front and rear half crankshaft bearing cheeks and both sides of the centre half crankshaft bearing cheeks.

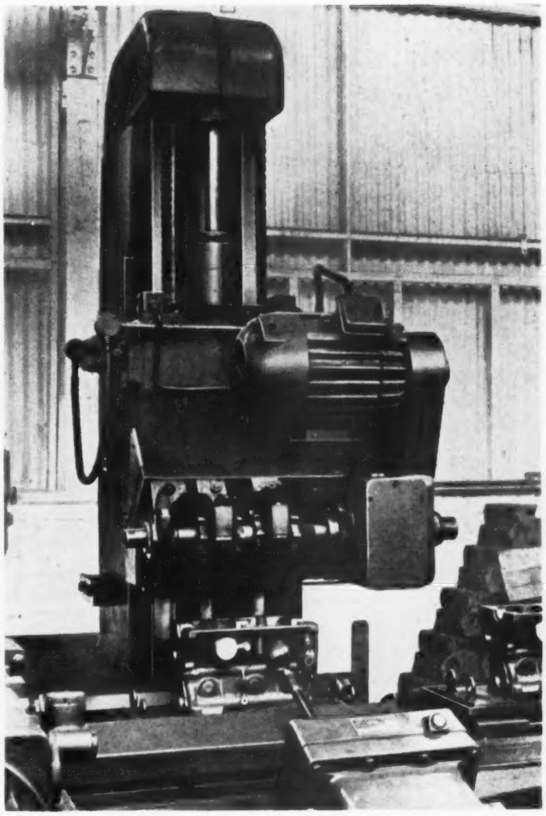


Fig. 8. Rise-and-fall milling head for machining the main bearing cheeks on the Austin 'A30' cylinder block

Drill units are used at the second working station. A horizontal unit at the left is tooled to open out the oil release valve hole to two diameters and depths by means of a stepped drill, and two other holes are drilled at the same time. The right-hand unit is angular and drills holes from the three main bearings to the main oil gallery.

Vertical and horizontal drill units are used at the third working station. The horizontal has three spindles tooled for opening out the oil release valve hole, spot facing the oil release valve boss and chamfering the large oil filter connection hole. The vertical unit drills a hole through into the oil release

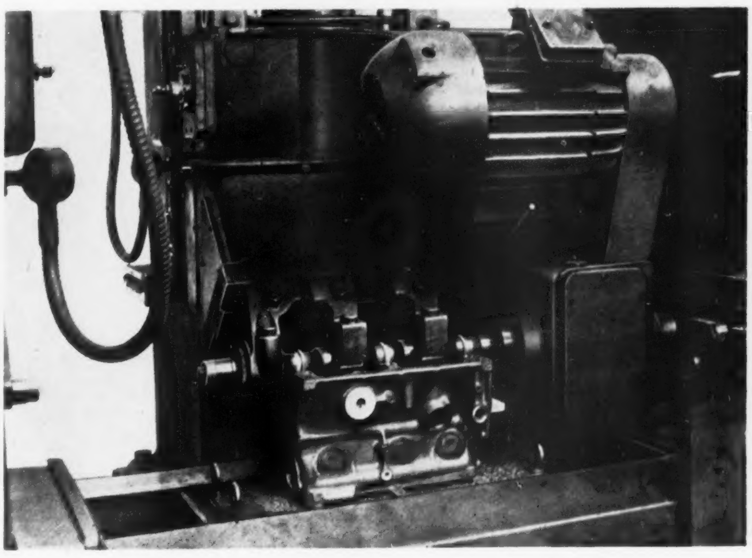


Fig. 9. Milling the bearing oil grooves on the machine shown in Fig. 7

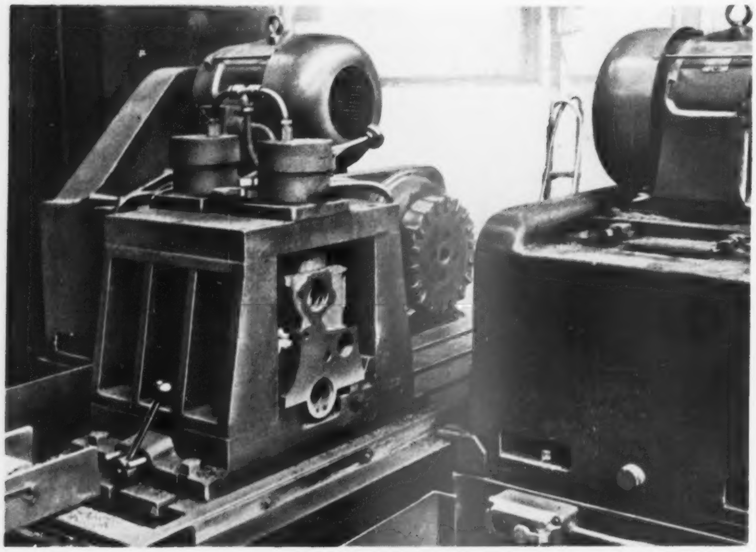


Fig. 10. Duplex miller for machining the end faces

valve hole, the oil suction hole and the oil filter drain hole.

At the next station, a two-spindle horizontal unit on the left-hand side carries a formed reamer for reaming the oil release valve bore to a specified depth and forming a 30 deg seating, and a drill for drilling through into the oil release valve hole. On the right-hand side of the line there is a blow-out and gauging unit of the type already described. Five blind sump screw holes and six bearing cap bolt holes are blown-out and checked for depth. There is also a horizontal blow-out unit on the left-hand side at the fifth working station. It is used for blowing-out and checking for depth the horizontal tapping holes on the off side of the cylinder block. The right-hand unit at this station is a 12-spindle vertical tapper for tapping the bearing cap bolt holes, five sump screw holes and an oil suction hole.

Both units are used for tapping at the next station. One taps five holes in the off side, while the other taps nine right-through sump screw holes. There is only a single angular drilling unit at the seventh working station. It is tooled to drill three oil holes, one from the front crankshaft bearing to the front camshaft bearing, one from the centre crankshaft bearing to the centre camshaft bearing, and one from the oil filter hole to the main oil gallery. Two single spindle units are used at the next station. At the left-hand side an angular unit is used for reaming the oil hole from the centre crankshaft bearing to the centre camshaft bearing, while a vertical unit at the right drills a hole from the sump face to the oil release valve hole.

Only one unit is mounted at the ninth working station. It has a rise-and-fall milling head, see Fig. 9, tooled for machining an oil groove in each of the three crankshaft bearings. To complete the operations on this machine, there is a similar rise-and-fall milling head at the tenth working station. It mills a bearing tab slot in each of the three crankshaft bearings.

At this stage the sump face is finish milled. Adjacent to the milling machine there is a receiver gauge for checking the release valve hole, the angular oil holes, the thrust faces, the distributor bores, the oil grooves and the bearing tab slots. Immediately after the sump face is finish milled, the front main bearing cap is fitted, the block is turned over on to the sump face and loaded into an Archdale duplex milling machine, Fig. 10, in which the front and rear ends are finish milled.

Further transfer machining is then carried out on an Archdale seven-station in-line machine, Fig. 11. The casting is mounted on the platen in the same manner as that used at the first transfer machine in the cylinder block line, but with its longitudinal axis at

Fig. 11. Archdale seven-station in-line transfer machine

right angles to the longitudinal axis of the machine. At the first working station on this machine, 10 holes are drilled in the rear end and the end of the main oil gallery is reamed for a depth of ½ in from a horizontal unit at the left, while another horizontal unit at the right is used to drill nine holes in the front end, and the other end of the main oil gallery is reamed. Horizontal multi-drill units are also used at the next station, one to drill 13 holes in the rear end and the other to drill seven holes in the front end.

Many of the holes drilled at the first two working stations on this machine have to be tapped. To prepare them for the tapping operations, there are two horizontal blow-out and checking units at the third working station. Six holes are tapped in the front end and 15 in the rear end at the next station. At the final working station the left-hand unit is a 12-spindle horizontal tapper for tapping the rest of the holes in the rear end, while the right-hand unit is a four-spindle vertical borer, see Fig. 12. Each boring bar carries two tools, one for chamfering the top and the other for chamfering the bottom of the cylinder bores. It is almost unnecessary to say that the bottom chamfering tools are of the tangential-feed type.

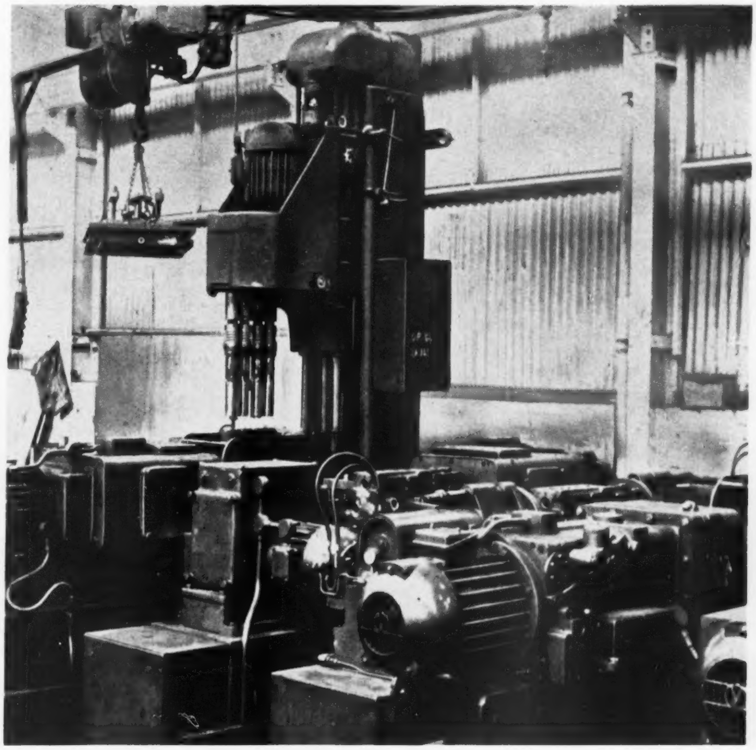


Fig. 12. Four-spindle boring head for chamfering the cylinder bores, top and bottom

From the transfer machine, the block is passed in sequence to two Archdale four-spindle drilling machines. On the first, the eight tappet guide holes drilled early in the machining sequence are opened out, and on the second they are reamed to size. Each of these machines incorporates a three-station indexing table of the type described earlier for the first tappet guide hole drilling operation. The reaming machine is shown in Fig. 13.

The centre and rear main bearing caps and the rear cover are then fitted, before the block is transferred to an Archdale two-way borer, on which machine the crankshaft bearings and the camshaft bearings are semi-finish bored. From a rear angular head the distributor housing bore is semi-finished and the distributor shaft bore is opened out and chamfered. The block then passes to another Archdale machine that differs only in having a third head at the left-hand side. On this machine the crankshaft bearing and camshaft bearing bores are finished to size from the right-hand head, while the distributor housing bore is finish machined and distributor shaft hole is reamed from the rear angular head. The left-hand head is used to ream two dowel holes in the rear end.

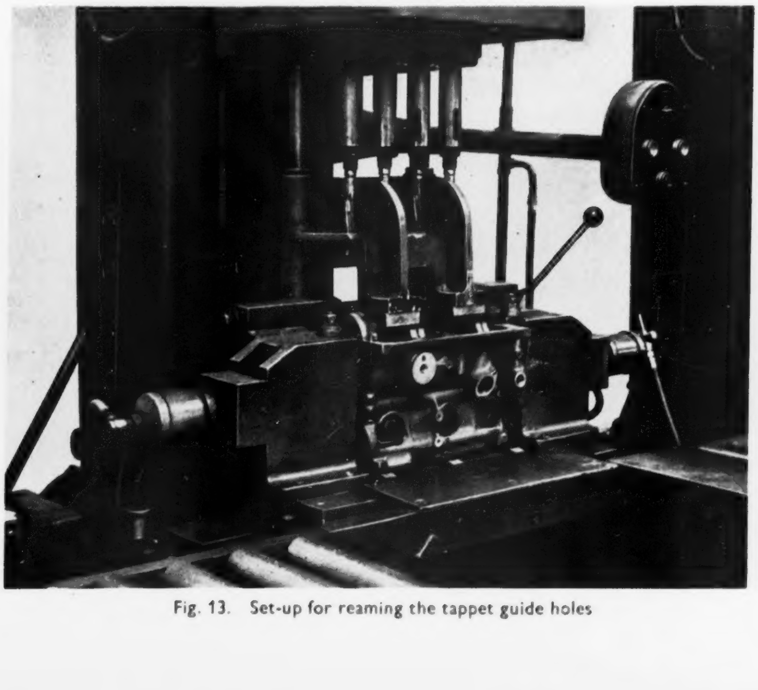


Fig. 13. Set-up for reaming the tappet guide holes

Only one further machining sequence calls for comment. It is carried out on an Archdale nine-station in-line transfer machine, Figs. 14 and 15, for finishing the cylinder bores. The practice at this machine varies from that employed at the other transfer machines in the cylinder block line, in that two blocks are mounted side-by-side on the platen. At each of the working stations there is a single vertical boring unit; the units are arranged alternately to the left and to the right of the transfer line.

The sequence on this machine is such that one block is machined at the first, third and fifth working stations while the other block is machined at the second, fourth and sixth stations. The final station is common to both blocks. At the first station a four-spindle unit semi-finishes the cylinder bores of the left-hand block; a similar unit carries out the same operation on the right-hand block at the second station. Finish boring is carried out at two stations for each block. At the third station Nos. 1 and 3 cylinders of one block are fine bored and Nos. 1 and 3 cylinders of the other block at the fourth station. Nos. 2 and 4 cylinders are fine bored in one block at the fifth station and at the sixth

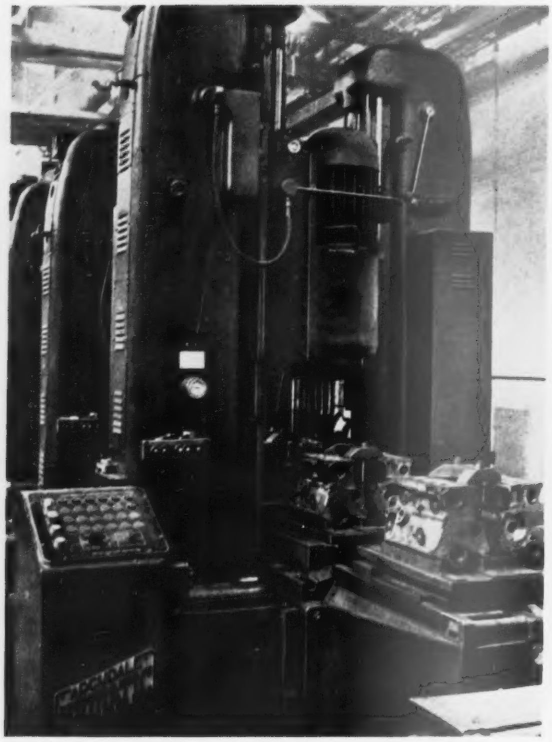


Fig. 14. Archdale nine-station in-line transfer machine for semi-finish and finish machining the cylinder bores

station for the other block. At the final working station, wire brushes are mounted on boring bars in an eight-spindle vertical head for finish brushing the cylinder bores. Every cylinder bore is inspected for finish and then graded for size by means of Solex pneumatic equipment. The machining is then completed by finish milling the top face.

It is not intended to describe in detail the machining of the cylinder head, but attention must be drawn to the small amount of handling that is necessary. This is chiefly due to the introduction of two automatic in-line transfer machines into the cylinder head machining line. They are both Archdale machines. The sequence will be described very briefly.

The first operation is carried out on a vertical two spindle milling machine with two work fixtures. In the first fixture the top face is machined; the casting is then transferred to the second fixture with the joint face up for machining. The casting is then transferred to a three-way unit machine with a three-spindle vertical and a single-spindle horizontal head for drilling certain holes, and a horizontal head for milling the manifold face.

From the three-way unit machine, the casting passes to a six-station in-line transfer machine. This machine has drill units at both the right and the left of the transfer line at each working station. On leaving the first transfer machine the casting is viewed and then passed to a 19 station-in-line transfer machine, shown in Fig. 16. By the time the casting reaches the end of this machine, all the necessary drilling, reaming and tapping have been carried out.

Only two further machining operations are carried out on the casting. At the first, the joint face is finish machined on a conventional milling machine; at the second the valve guide holes, the valve throats and the valve seats are finish machined. Krause fine boring machines are used. This machine has two stations, one tooled for fine boring two exhaust valve guide holes and finish forming their valve throats and seats, and the other for carrying out similar operations on two inlet valve elements. After these four holes have been machined, the fixture is indexed to bring the other exhaust and inlet holes into the machining position and the machining cycle is repeated. In other words, the elements for the four exhaust valves are completed in two machine cycles; the casting is then transferred to the second fixture and the elements for the inlet valves are completed in another two machine cycles.

Crankshaft machining

Without any doubt the most interesting feature of the machining methods for crankshafts is the use of multi-station in-line transfer machines. No other company in this country uses transfer machines for this component, nor to the best of our knowledge have such machines been developed in America. Apart from the transfer machines, only the Wickes machines for rough and finish turning the journals call for comment; all the other machines are of conventional, well-known and proven types. The set-up for rough journal turning is shown in Fig. 17.

On the Wickes machine for rough turning, a turning tool on the rear tool slide turns the centre journal while two forming tools on the front slide machine the webs. One of the forming tools turns down the front fillet, turns down the web and blends into the journal; the other turns down the rear fillet and turns down the web. For machining the front journal, a turning tool and a face and turning tool are mounted on the rear tool slide and a forming tool and a turning tool on the front slide. The tools on the rear slide turn the front of the front journal, face down the front of the front journal and turn the rear of the gear diameter. Those on the front slide turn down the fillet at the front journal and blend it to the journal while the turning tool turns down the web.

For the rear journal and flange, two form turning tools are mounted on the rear tool slide and two turning tools on the front tool slide. The forming tools turn down the rear fillet, turn the journal, turn down the front face of the flange, turn rear of oil return thread and form a radius; turn down the rear face of the flange and turn the register. The front tools turn down the front fillet, turn a step on the web and turn

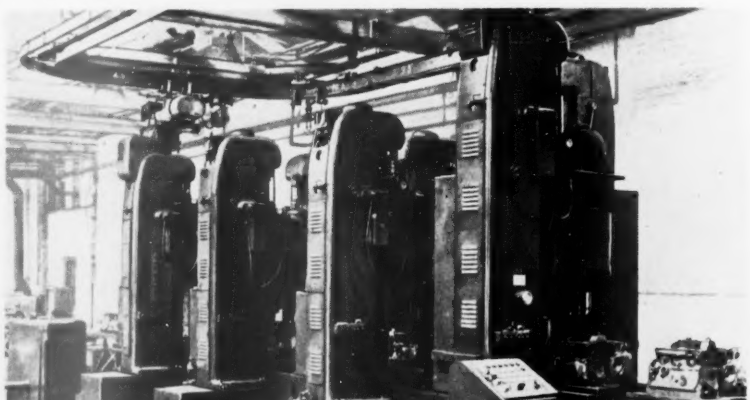


Fig. 15. The loading station for the machine shown in Fig. 14

Fig. 16. 19-station in-line transfer machine for the Austin 'A30' cylinder head

the front of the journal. A somewhat similar complex operation cycle is carried out on the Wickes machine tooled for finish turning the journals.

The first of the transfer machines in the crankshaft line is a seven station machine with Austin unit heads. It is tooled for drilling, tapping and forming both ends of the shaft. The loading station is shown in Fig. 18 and a general view of the machine in Fig. 19. Location in the work fixture is taken from turned journals and from location pads milled at an earlier operation.

At the first working station the left-hand head is a horizontal drill unit with a five-spindle multi-head. It drills a central hole and four tapping holes in the flange. The right-hand head at this station drills a central hole in the other end of the shaft. Similar units are mounted at the second working station. At this station the left-hand head is tooled to continue drilling the central hole in the flange to a smaller diameter and to a depth of $2\frac{1}{16}$ in. Coning drills are mounted on the other spindles of the five-spindle head for chamfering the flange holes. At the same time the right-hand head opens out the hole in the other end of the shaft for a depth of $\frac{1}{4}$ in.

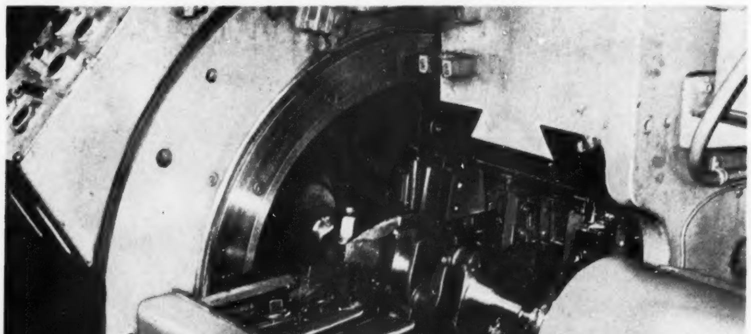


Fig. 17. Set-up for rough turning crankshaft journals on a Wickes journal turner

Horizontal units with single tools are used for both heads at the next station. The left-hand head carries a form tool with an inserted Galtona blade for forming a 50 deg angle on the flange end while the right-hand head is tooled for facing the end to length. Two horizontal tapping units are used at the fourth working station. A four-spindle tapping head is mounted on the left-hand unit. It taps four holes in the flange. The right-hand head taps the central hole in the rear end of the shaft. At the final working station there is a single coning tool on each head.

The second transfer machine has 11 stations, nine working and one each for loading and unloading. It is tooled for countersinking and drilling the oil holes from the journals to the crank-pins, and for reaming the holes for restrictors. To carry out these operations, it is necessary to change the angular disposition of the shaft in relation to the heads at certain stations. To obtain the angular changes, a special indexing work fixture is used, see Fig. 20. Cams on the machine effect the indexing at the appropriate stations.

At the first working station the left-hand head is a horizontal drill unit with a three-spindle attachment. It countersinks Nos. 1 and 4 crankpins and the centre journal. The right-hand head has a four-spindle head attached to a horizontal drill unit for countersinking Nos. 1 and 3 journals and Nos. 2 and 3 crankpins. At the second station the shaft is indexed to a position for drilling oil holes. At this station

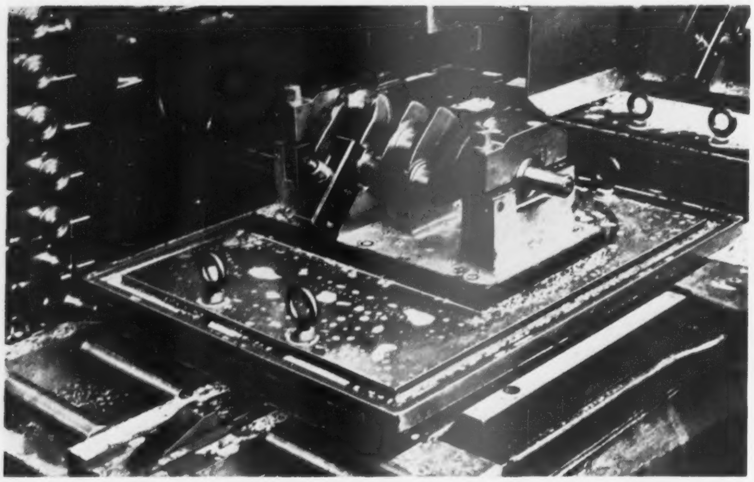


Fig. 18. Loading station for the transfer machine shown in Fig. 19

the left-hand head is a horizontal reciprocating drill unit with a spade point extension drill for drilling a $\frac{3}{16}$ in hole from No. 4 crankpin to No. 3 journal, $1\frac{1}{4}$ in deep. The right-hand head has a horizontal reciprocating unit with a two-spindle attachment. It carries spade drills for drilling a hole from No. 3 journal to No. 4 crankpin and a hole from No. 3 crankpin to the centre journal. Each hole is drilled to a depth of $1\frac{1}{4}$ in.

The shaft disposition remains the same at the next station where a horizontal drill unit at the left is used to ream the hole at No. 4 crankpin to a depth of $\frac{1}{2}$ in. A two-spindle head is mounted on the horizontal drill unit at the right. One spindle reams the hole at No. 3 crankpin to a depth of $\frac{1}{2}$ in, while the other carries an extension drill that completes the drilling of the hole from No. 3 journal to No. 4 crankpin. This drill is $\frac{1}{64}$ in less in diameter than the drills used at the previous station for starting the hole from No. 3 journal to No. 4 crankpin.

Only one head is used at the fourth working station. It is a horizontal drill unit at the right hand for continuing drilling the hole from No. 3 crankpin to the centre journal for a further $1\frac{1}{4}$ in but at $\frac{1}{64}$ in smaller diameter. At the next station also, only a right-hand head is used. It is a horizontal drill unit that completes the drilling through from No. 3 crankpin to the centre journal.

At the sixth working station the fixture is automatically turned through an angle to bring Nos. 1 and 2 crankpins into position for drilling. From the left-hand head a horizontal drill unit drills a $\frac{3}{16}$ in diameter hole $1\frac{1}{4}$ in deep from No. 1 crankpin to No. 1 journal, while a two-spindle head on a horizontal drill unit at the right drills $\frac{3}{16}$ in diameter holes $1\frac{1}{4}$ in deep from No. 1 journal to No. 1 crankpin and from No. 2 crankpin to the centre journal.

With the shaft in the same angular disposition, at the next station the left-hand head is used to ream the hole at No. 1 crankpin for a depth of $\frac{1}{2}$ in, while the right-hand head reams the hole at No. 2 crankpin and drills through the hole to No. 1 journal. At the eighth working station the hole from No. 2 crankpin to the centre journal is drilled for a further $1\frac{1}{4}$ in from the right-hand head, and at the ninth station this hole is drilled through.

Connecting rod machining

Rods and caps are produced from separate stampings. The vertical axes of the rods are offset in relation to the central vertical plane of the big end, the rods for Nos. 1 and 3 cylinders being offset in one direction, while those for Nos. 2 and 4 cylinders are offset an equal amount in the opposite direction. In other words, two right- and two left-hand rods are required for each engine.

The machining sequence will not be described in full; only the more interesting operations will be discussed in detail. Two preliminary operations are carried out to give locations for

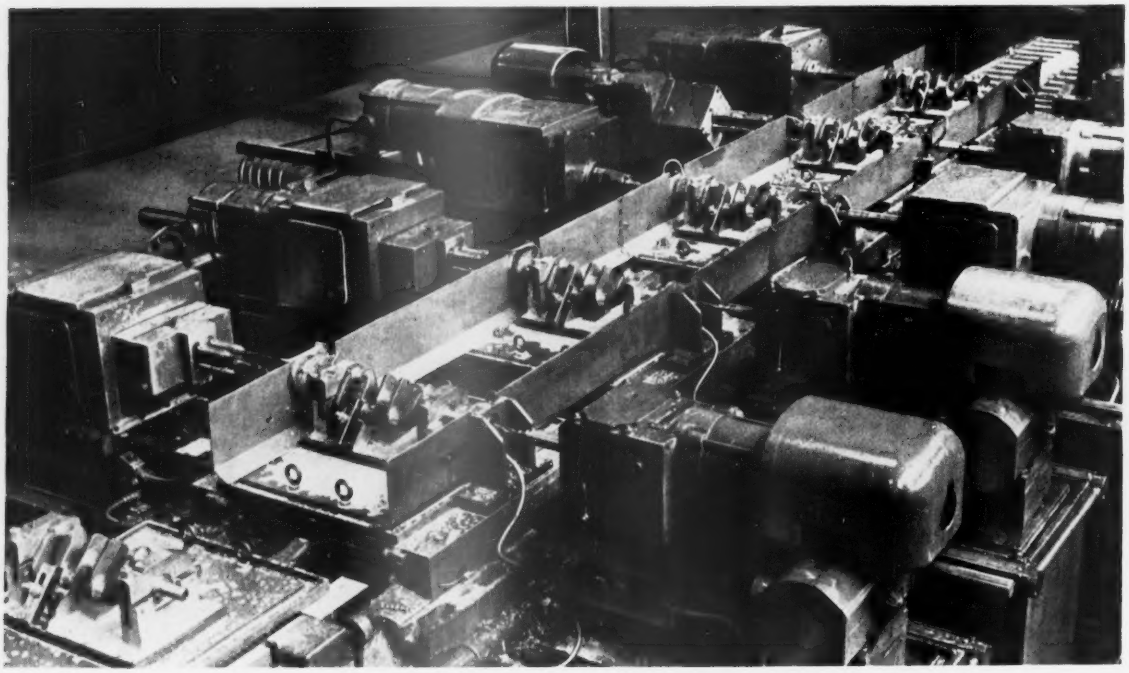


Fig. 19. Seven-station transfer machine for drilling, tapping and forming the crankshaft ends

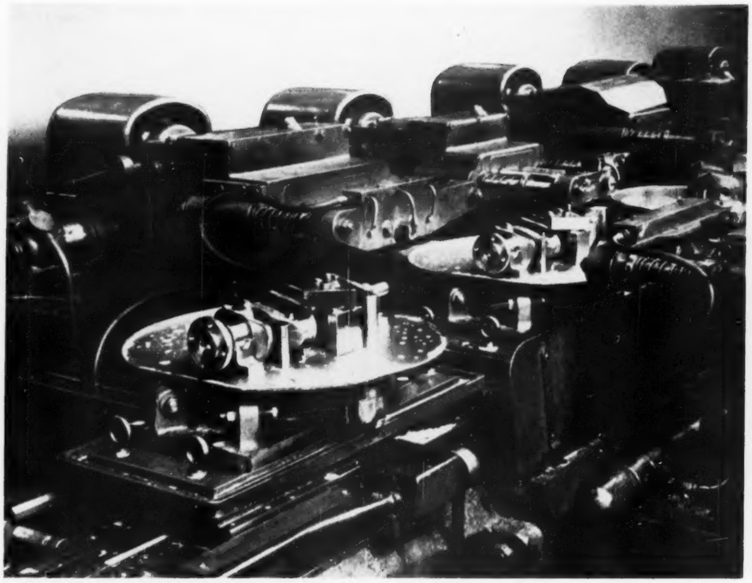


Fig. 20. Loading station and first working stations on the transfer machine for drilling and reaming oil holes from the crankshaft main journals to the crankpins

subsequent operations. At the first of these, the rod is rough ground on both side faces and at the second the gudgeon pin hole is drilled and reamed. After these location points are machined, the rods are loaded two at a time, one right- and one left-hand, into a fixture on a Cincinnati Vertical duplex surface broaching machine. This machine is tooled to broach the crankpin bore, the joint face and register and form a step.

From the broaching machine the rods are transferred to an Archdale 11-station in-line transfer machine, Figs. 21 and 22. The work-holding fixtures for this machine are arranged to hold two right-hand and two left-hand rods to give a balanced production in sets. At the first working station, a horizontal four-spindle drill unit at the left drills a tapping hole through the gudgeon pin boss of each rod, while a plunge cut horizontal milling unit at the right mills the bearing retaining slots in the big end half bores of the right-hand rods. Only one head is used at the next station. This is at the right of the transfer line, and is a plunge cut horizontal milling head for milling the bearing retaining slots in the two left-hand rods.

Two heads are used at the third working station. That at the right is a four spindle horizontal unit for drilling an oil hole, $\frac{1}{8}$ in diameter × $\frac{1}{8}$ in deep in each of the big ends, while the left-hand head is a 12-spindle angular drilling unit. It drills two bolt holes and counterbores the tapping hole in the gudgeon pin boss on each of the four rods. Only one head is used at the next station. It is an eight-spindle horizontal drill unit tooled to open out the bolt holes in each of the rods.

At the fifth working station four bolt holes are counterbored to clear for tapping from a four spindle head on one side, while at the other side a 12-spindle horizontal drill unit is tooled to spot face a surface on the gudgeon pin boss and countersink two holes for tapping on each rod. Only a blow-out and gauge unit is mounted at the next station. It clears the swarf from and checks the depth of the tapping holes. At the seventh working station there is again only one head. It is a 12-spindle horizontal unit for tapping one blind bolt hole, one through bolt hole and a hole in the gudgeon pin boss on each rod. At the next station an angular drilling unit at the left drills the oil hole through in each big end, while the right-hand unit blows-out eight holes for final tapping. Only one head is used at the final working station. It finish taps the bolt holes in each rod.

Fig. 21. Eleven-station transfer machine for connecting rods

In-line transfer machining is also employed in the production of the connecting rod caps. Only three machines are used to produce a cap ready for assembly to the rod. At the first, the cap is milled to width, and at the second the bore, the face and other elements are machined on a duplex vertical surface broaching machine. From the broaching machine the caps are passed to an Archdale seven-station in-line transfer machine. The work-holding fixture is arranged to take three components.

At the first working station a six-spindle horizontal drill unit drills the bolt holes in each cap. These holes are opened out at the next station where once again only one head is employed. At the third working station a six spindle horizontal drill unit at the left is used for spot facing the bolt holes, while a plunge cut milling unit at the right mills the bearing retaining slot in each cap. The bolt holes are countersunk each end from right- and left-hand heads at the fourth working station, and are then reamed from a horizontal unit mounted to the right at the final working station.

The operation sequence for the assembled connecting rod may be dealt with briefly. After assembly the rods are ground on both sides on a Lumsden surface grinder. In an Archdale double-ended horizontal machine each side of the crankpin bore is chamfered. The fixture on this machine takes one right- and one left-hand rod to be machined simultaneously The gudgeon pin hole is then opened out on an Archdale machine with an indexing-table, and a fixture that takes two right- and two left-hand rods. There is a loading station and two working stations. At the first working station the holes are opened out from the size drilled on the separate rod and at the second they are reamed to a diametral tolerance of 0·0005 in.

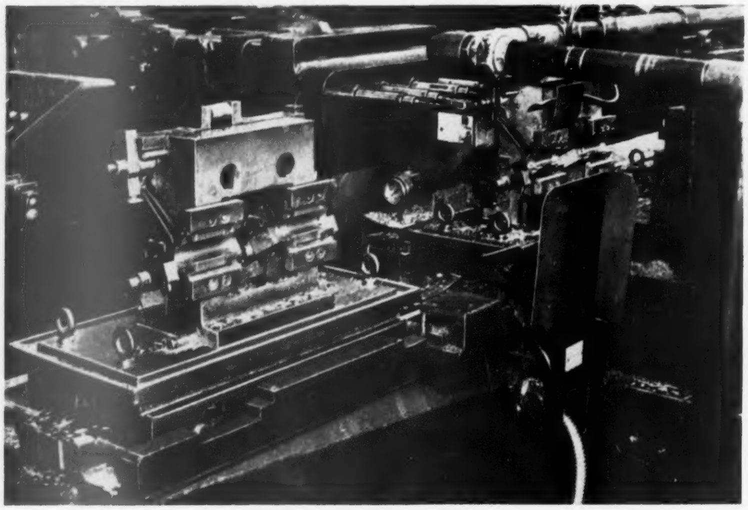


Fig. 22. The loading station for the machine shown in Fig. 21

A double-ended machine with a fixture to hold two components at each end is used for fine boring both the gudgeon pin hole and the crankpin bore. These elements are finished to size on a four-spindle turret honing machine that is arranged to hone two crankpin bores and two gudgeon pin bores simultaneously. The remaining few operations on the assembled connecting rod do not call for comment.

In the production of the other major engine component, the camshaft, there is no real break from conventional machining practice in so far as there is no use made of special machines. Standard machines are used throughout, but attention may be drawn to the manner in which multi-tool lathes are used to complete the machining with the least possible handling.

Engine assembly follows standard Austin practice. That is, the engine parts are marshalled in individual sets in the stores and are transferred to the assembly section on a conveyor, so that an operator never needs to leave his station to collect the parts he needs. A detailed description of the Austin assembly system and the methods for running in the engines was given in the *Automobile Engineer* for March 1947.

INSTITUTION OF MECHANICAL ENGINEERS

Forthcoming Meetings of the Automobile Division

The following meetings will be held during March:—

BIRMINGHAM CENTRE

Tuesday, 24th March, 6.45 p.m. General Meeting in the James Watt Memorial Hall, York House, Great Charles Street, Birmingham. Paper: "Some Problems Arising from the Wider Use of Small Diesel Engines," by J. H. Pitchford, M.A. (Cantab.), M.I.Mech.E.

LUTON CENTRE

Graduates' Section

Wednesday, 18th March, 7.30 p.m. Industrial Film Show at the Works of W. H. Allen Sons, Ltd., Bedford.

Thursday, 19th March. 7.30 p.m. Industrial Film Show at Luton Town Hall.

NORTH-EASTERN CENTRE

Wednesday, 18th March, 7.30 p.m. General Meeting in the Chemistry Lecture Theatre, The University, Leeds. Address by the Centre Chairman, Professor W. A. Tuplin, D.Sc., M.I.Mech.E.

NORTH-WESTERN CENTRE

Wednesday, 25th March, 6.30 p.m. Annual Dinner at The Engineers' Club, Albert Square, Manchester. Address by the Chairman of the Automobile Division, Mr. Maurice Platt, M.Sc. (Sheffield), M.I.Mech.E., entitled "The Changing Practice of Automobile Engineering."

SCOTTISH CENTRE

Monday, 16th March, 7.30 p.m. General Meeting in the Institution of Engineers and Shipbuilders, 39, Elmbank Crescent, Glasgow. Paper: "Research and the Engineering Process with Particular Reference to the Automobile Industry," by H. E. Merritt, M.B.E., D.Sc.(Eng.) (Lond.), M.I.Mech.E., (Member of Council).

WESTERN CENTRE

Thursday, 26th March, 6.45 p.m. General Meeting in the Grand Hotel, Bristol. Paper: "Research and the Engineering Process with Particular Reference to the Automobile Industry," by H. E. Merritt, M.B.E., D.Sc.(Eng.)(Lond.), M.I.Mech.E. (Member of Council).

The following meetings will be held during April:—

LONDON

Tuesday, 14th April, 5.30 p.m. General Meeting at Storey's Gate, St. James's Park, S.W.1. Paper: "The Jaguar Engine," by W. M. Heynes, M.I.Mech.E.

NORTH-EASTERN CENTRE

Wednesday, 15th April, 7.30 p.m. General Meeting in the Chemistry Lecture Theatre, The University, Leeds. Paper: "Research and the Engineering Process with Particular Reference to the Automobile Industry," by H. E. Merritt, M.B.E., D.Sc.(Eng.)(Lond.), M.I.Mech.E. (Member of Council).

RECENT PUBLICATIONS

Brief Reviews of Current Technical Books

Climbing Ability and Power Output (Bergsteigfähigkeit und Literleistung)

By Dr.Ing. Wolfgang Flössel. Stuttgart: FRANKH'SCHE VERLAGSHANDLUNG. $9\frac{1}{2} \times 6\frac{1}{2}$. 284 pp. 157 illustrations. Price DM 25 (1 DM = 1s. 8d.).

The interaction of engine power output characteristics with car performance requirements, the possibilities offered to meet these by alterations to engine design, the limits set by the necessity to ensure low fuel consumption, are important problems dealt with not only on engineering but general policy levels. But how many engineers have closely considered all the variables and their interaction, let alone written a book on this subject?

This book deals with this very matter in a manner which is clear, practically all embracing and attractive, not only to the theorist and the practising engineer, but to the technically appreciative layman as well.

The author deals briefly with the various definitions of engine flexibility (which he calls elasticity E) and then puts forward his own according to which $E = (T_2/T_1)(n_1/n_2)$ where T_1 = torque at maximum power output, T_2 = maximum torque and n_1 and n_2 the respective speeds. Next he deals with the influence of various factors on the shape of the torque curves of petrol engines and on their losses. This is followed by an analysis of flexibility and the numerous factors affecting it, a subject of great importance and absorbing interest to engine designers. The development of flexibility characteristics since 1905 is considered in the light of the requirements stipulated throughout the years by various authorities. Some of these requirements were conflicting and a few not altogether rational. The flexibility of rotary valve engines is dealt with next, and these are followed by supercharged and two-stroke engines, altogether some 83 pages being devoted to petrol engines.

The flexibility of diesel engines is analysed along somewhat similar lines, the performance data of some 170 engines forming the basis for considering the various factors affecting the all-important E, which is then compared with that of petrol engines. Having thus dealt with E the author logically proceeds to deal with the E requirements of the vehicles and the possibility of improving vehicle flexibility by suitable design. The effect of E on the ability to overcome tractive resistance (climbing ability) and acceleration is thoroughly dealt with, together with the performance ensured by the use of oversize engines. Since considerations of this kind must be obviously based on a good knowledge of tractive resistance data, some ten pages are devoted to this somewhat difficult subject, whilst the next ten pages deal with the derivation and application of the flexibility function $\phi = (T_2/T_1) - [1/(n_1/n_2)^2]$.

It is gratifying to note that the author has not neglected the motor cycle, the "climbing ability" of which is considered in a similarly thorough and thought-provoking manner. The conclusions derived from these investigations occupy the last eight pages, whilst an appendix dealing with the engine performance of a number of representative vehicles concludes the first part of the book, which has 182 pages, and which was submitted as thesis for the degree of Doctor of Engineering at the Technical High School, Stuttgart.

It is only natural that automobile engineers should be interested in the performance of other modes of transportation, and to meet and encourage this demand the second part of the book deals with the flexibility of road and rail vehicles powered by I.C. engines, steam and electricity. As before, the author deals with the tractive resistance—this time of rail vehicles—in considerable detail and then proceeds to analyse the flexibility of modern steam locomotives, steam turbine locomotives and D.C. and A.C. (single phase and three phase) locomotives before reverting to I.C. engined rail vehicles provided with mechanical, electric or hydraulic transmissions. The flexibility of road vehicles powered by steam or deriving power from storage batteries does not escape the author's attention, nor is the trolleybus forgotten, either.

The final chapters deal with overall considerations of ϕ, the effect of modern hydraulic transmissions on flexibility and the interdependence of power output per unit swept volume (Literleistung) and flexibility. Some 25 pages are devoted to tables dealing with engine and car data. They include engine flexibility of modern passenger, racing car and lorry engines, tractive resistance data and E and ϕ data for a considerable number of motor cycle and car engines. The bibliography embraces 133 references.

The knowledge, erudition and industry displayed and the careful study which the author has devoted to the subject will be apparent from the brief review of this excellent and well produced book. It must be said that this illuminating and thought-provoking book at last closes an important gap in the current technical literature. It should be studied by engineers who have to deal with the design of engines and vehicles and executives who must make decisions affecting their design and operation.

Vibration Dynamics of Fast Road Vehicles (Schwingungsdynamik des Schnellen Strassenfahrzeugs)

By Prof. Dr.-Ing. E. Marguard. Essen: VERLAG W. GIRARDET. 1952. $6 \times 8\frac{1}{2}$. 228 pp. Price DM 20.

Even the title will commend this book to automobile engineers and they will not be disappointed by its contents. The author, Professor at the Technical High School at Aachen, has endeavoured to improve the art of vehicle design by providing the theoretical background for the appreciation of vehicle behaviour pattern to give the engineer a useful tool for rational pre-designing of vehicles. In this task he succeeds admirably.

The *raison d'être* of the book is given in the introduction. "It is particularly important to a poor country that mental and paper work is far less expensive in materials and money than, for example, the American methods of trial and error. Present day conditions are similar to those of the years after 1919; then, too, it was only possible for the industrially poorer countries to hold their own in the face of great financial and experimental facilities of America by a concentrated mental effort. Today and tomorrow conditions will remain the same."

The general problems faced by vehicle designers are dealt with to provide the necessary foundation. This embraces forced vibrations, road obstacles and an elucidation of the effect of mathematical simplification on the accuracy of final determinations. Since the evaluation of vibration dynamics of particular vehicles requires the knowledge of basic vehicle data, these are dealt with next. Details are given of experimental determination of the location of the centre of gravity, tyre and spring (both rubber and steel) constants and shock absorber performance data. The theory and practical aspects of damped vibrations as encountered with automobiles is discussed in considerable detail and this is followed by a chapter on the theory and practice of the determination of the moment of inertia of components and complete vehicles. A 20-page chapter is devoted to the single degree of freedom system, followed by some 30 pages dealing with two degrees of freedom, always with particular reference to automobiles.

Having devoted half of the book to a thorough treatment of fundamental considerations, the author deals in the remaining half with natural frequencies and resonance ranges of vehicles in three planes, and the vibration modes due to passage over various types of obstacles. A separate chapter is devoted to the landing shock experienced by aircraft, which at this stage of its progress comes under the heading of the book.

The research problems encountered in connection with vehicle vibrations are dealt with in considerable detail. Since road and rig tests are of great importance, the variables involved are discussed, together with the design of suitable rigs and the testing of scale models. The theoretical background of shimmy as developed by de Lavaud, Becker, Fromm and Maruhn are next dealt with in some detail, and this is followed by a chapter on curve passage dynamics and the rolling stability of aeroplanes and road vehicles generally. The book concludes with an appendix on the approximate integration of differential equations.

The value of the book is enhanced by numerous worked out examples and the very clear diagrams, and it should be of great value to all dealing with the important subject of vehicle dynamics. The value of the book would be extended still further by devoting some space to electro-mechanical analogies, which are very useful in the study of complex vibration problems, and also by extending the bibliography to the many important papers published on the subject matter of vehicle vibrations outside Germany. However, this in no way detracts from the value of this excellent book.

HYDRAULIC VALVE TAPPETS

A Review of Current Practice

by R. Brown-Waring

AN outstanding difference between American and British practice is the general adoption of hydraulic valve tappets on American engines and their apparent lack of favour with British designers. This is peculiar, since in experiment and practice hydraulic valve tappets have shown marked advantages that cannot be obtained by other means, particularly on vee and overhead valve engines.

Modern developments have led to the increased use of such engines, especially in the U.S.A., and a factor of great importance in their design is that of valve operation and tappet adjustment. This presents a difficult problem, particularly in vee type engines, not merely on account of the number of valves but also because it is virtually impossible to provide simple means of accessibility.

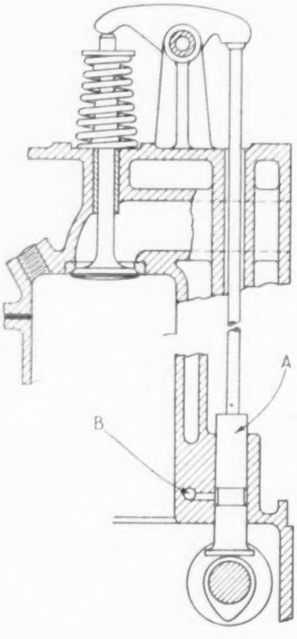
For a variety of reasons, this question has recently assumed some importance on English vehicles with engines of diverse construction and emplacement on which valve tappet adjustment is a very difficult operation. An obvious solution is to eliminate the need for attention; this, in fact, is one fundamental reason for the use of hydraulic valve tappets. Other technical advantages are so many and so progressive that it is difficult to envisage how hydraulic valve tappet operation can be omitted from any conventional engine assembly in the near future. Their incorporation in the design does not entail a multiplicity of parts, nor an appreciable increase in production costs.

Development has already been sponsored by Rolls - Royce, Cadillac, Packard, Chrysler and other motor manufacturers on designs with self-adjusting valve tappets of the oil pressure type. In these, the pressure is applied behind the tappet end to maintain contact with the valve stem during actuation. When the instant of contact passes, a port is uncovered to reduce the pressure sufficiently to provide a minimum clearance to ensure that the valve definitely finds its seat.

In addition to this important feature, hydraulic valve tappets give other advantages, which may be summarized as follows:—

1. Automatic adjustment.
2. Accurate valve timing.
3. Improved engine performance.
4. Quiet operation.
5. Increased durability.

Considerable ingenuity in design has been exercised to attain these features in specific engines.



A = hydraulic tappet B = Engine lubrication system

Fig. 1. Hydraulic tappet on overhead valve engine

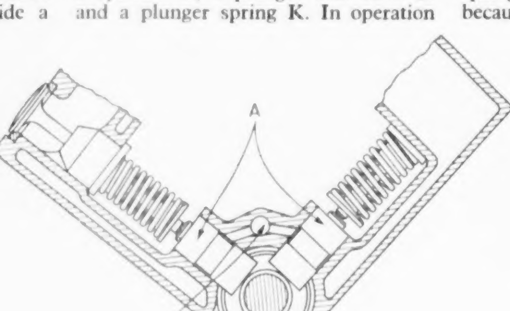
To enable the exact function of the hydraulic tappet to be appreciated, Figs. 1 and 2 show typical arrangements for overhead valve and vee type engines. These illustrations make it clear that no particular modifications are involved in the adoption of hydraulic tappets. Their simplicity is exemplified by the Zero-lash device which is shown in Fig. 3.

In the Zero-lash, oil under pressure from the engine lubrication system is fed into the tappet assembly A via the hole H. The hydraulic unit comprises a cylinder B, a plunger ball valve D and a plunger spring K. In operation with the face of the tappet on the base circle and the engine valve on its seating, the spring K lifts the plunger C so that the upper end of the plunger makes contact with the valve stem. This eliminates any clearance whatever in the operating mechanism.

As the plunger moves upwards, the volume of the pressure chamber L is increased, the fluid pressure moves the ball valve D from its seat, and the chamber L is completely filled with oil lower half of the tappet. As the camshaft rotates, the cam forces the tappet A upwards, tending thereby to decrease the volume of the chamber L and forcing the ball valve on to its seat. Further movement of the cam will continue to force the tappet upwards, while the oil confined in chamber L acts as a member in the from the supply chamber J in the operating gears; in fact, the engine valve is lifted on a column of oil.

So long as the engine valve is off its seat, the column of oil sustains its load, and there is a predetermined leakage of oil between the plunger C and the chamber L. This is necessary to provide room for the oil required to fill the chamber as soon as the tappet leaves the receding flank of the cam to contact the base circle. This maintains adjustment.

It should be noted that it is not possible for this hydraulic tappet to hold the engine valve open when it should normally be closed. When the tappet is on the base circle of the cam, the principal force tending to hold the valve open is the pressure exerted by the spring K, which is much lighter than the force exerted by the spring that closes the engine valve. An additional force tending to push the plunger upwards is that due to the pressure of the oil supply to the chamber J formed in the tappet. The effect of this is very slight, since an oil pressure of 500 lb/in^2 is required to overcome 100 lb spring pressure. This relationship exists because of the small area offered by the plunger.



A = hydraulic tappets B = engine lubrication system

Fig. 2. Hydraulic valve tappets on vee type engine

Another design of hydraulic tappet that gives automatic adjustment is illustrated in Fig. 4. In this design the thrust is transmitted through a hydraulic cushion, but a clearance setting device is also introduced. The engine cam is shown at A and the valve stem at B; the cam acts on a cylindrical member C which is associated with a cup D that forms the hydraulic chamber. There is a clearance hole G in the upper portion of D. Through

the clearance hole there passes a reduced portion of the thrust member F in contact with the valve stem B. A sleeve, which is a press fit on the end of this thrust member, has a flange that forms an abutment for the spring E. Relative movement of the thrust member F and the cup D causes the flange to seal the clearance hole at the top of D, and so completely closes the chamber formed by C and D, and which is full of oil.

The cycle of operations begins when the cam A starts lifting. Cylinder C,

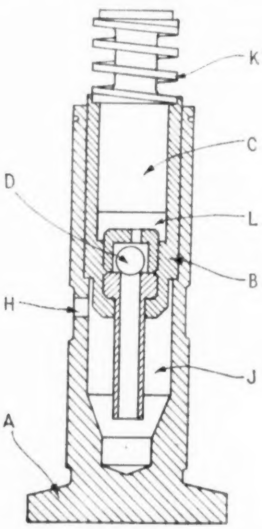


Fig. 3. Zero-lash hydraulic tappet

and with it the cup D, is raised, but the thrust member F is held against any upward movement by the valve spring H. This closes the clearance hole, and from then on the oil in the main chamber acts as a solid abutment to transmit thrust. As soon as the engine valve becomes seated and the valve spring becomes inoperative, the spring E comes into action to force the cup C in a downward direction on to the base circle of the cam. This opens the clearance hole G. If any leakage of oil has occurred, or if any adjustment is necessary to compensate for temperature change, oil is drawn into the chamber to restore the original conditions ready for the next cycle. It will be noted that the clearance in the valve system may be set at any desired amount by adjustment of the distance at the cap at G. In addition, the construction is such that should all the oil be lost—a most remote possibility—the valves would still be operated.

A minor problem with all hydraulic tappets is the maintenance of oil within the tappet, since a very slight leakage will affect the instantaneous action that is so essential to efficient valve actuation. Many devices have been introduced for this purpose. An outstanding example is the Weatherhead (U.S.A.) hydraulic valve tappet shown in Fig. 5.

In this, the valve tappet comprises a lower cylinder B with a hardened face riding on the cam A, and a hollow piston D closed by a hardened cap that abuts on to the valve stem H. A suitable relatively incompressible fluid fills the cylinder and a reservoir formed by a resilient sac C carried in the hollow piston. The sac is under the constraint of a spring G located between the piston cap and a pressure plate seating on top of the sac.

Communication between the cylinders is by means of a metered orifice in the head of the piston D. To give the required pressure and for ease of manufacture, a relatively large hole in the piston head is partly closed by a cylindrical plug E, which has a ground flat along one side to leave a small orifice of the required sectional area. The piston may be seated in the cylinder by means of a ring F of rubber or synthetic material unaffected by oil or spirit, or alternatively by means of a continuous sac with a beaded mouth adapted to snap into an annular groove formed in the reduced end of the piston.

The Cadillac scheme for hydraulic tappets as used on V8 engines is illustrated diagrammatically in Fig. 6. These units are arranged in blocks of four. They are all supplied under pressure to the reservoir E, wherefrom any bubbles may escape through the diagonal upper hole G past the tappet guide H into the crankcase. Through the lower hole F, oil is led *via* the vertical passage D into an annular groove J in the lifter body C. Thence it goes *via* a further hole into the valve lifter.

As the peak of the engine cam A comes into contact with the tappet B, the hydraulic plunger K is actuated upwards by the springs L to take up any clearance between the valve stem M and the cam. As the cam revolves, the initial pressure developed seats the ball valve N, thus trapping oil beneath the plunger. The valve is then lifted

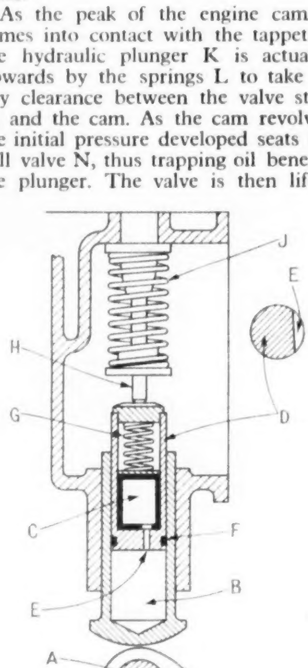


Fig. 5. Weatherhead (U.S.A.) hydraulic tappet

on a column of oil. While the valve is off its seat, a slight leakage of oil occurs to compensate for any expansion in the complete valve mechanism. The chamber beneath the plunger is replenished with oil as the valve closes; this eliminates clearance for the next cycle.

A different approach to design involving automatic tappet adjustment has been employed for the Packard 12 engine. In this instance the hydraulic mechanism is constructed as a unit separate from the valve tappet. It is shown in Fig. 7. The engine cam operates a tappet rocker H through the medium of a roller D. This roller is

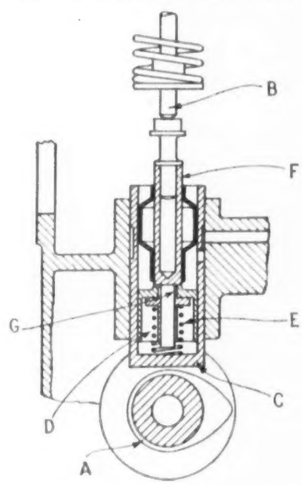


Fig. 4. General Motors hydraulic tappet

eccentrically mounted, and has a lip formed near its pivoting point to make contact with an extension piece of the piston B, which operates in a cylinder attached to the crankcase.

Movement of the piston is controlled by a compression spring K housed within it. The spring tends to press outwards, while motion in the reverse direction is resisted by the oil in the inner chamber. Since the oil intake valve V is mounted in the stationary member, movement of the piston does not cause sufficient depression to force the valve from its seat. For this reason a dashpot arrangement is incorporated.

Oil is fed under pressure to the reservoir, from whence it flows to the cylinder by gravity only. To ensure correct functioning, close limits are necessary between the piston and the cylinder. The permissible tolerance is only plus or minus 0·0001 in. If the oil leak be at too great a rate, there is noisy operation. On the other hand, if leakage is too slow, the engine valve J would be off its seat for too long a time after passing through a period of valve spring disturbance. This mechanism has already proved highly satisfactory. By its originality and obvious advantages, this construction offers possibility for further progress.

It is perhaps almost unnecessary to state that in any hydraulic valve tappet application, cleanliness of oil is

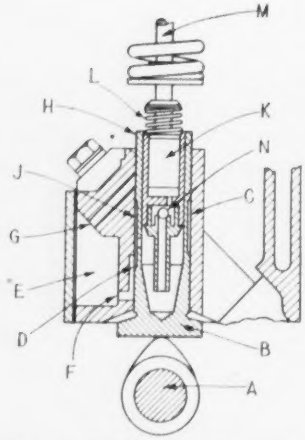


Fig. 6. Cadillac hydraulic tappet

important. It is essential that all possible means be adopted to prevent the entry of dirt or air into the adjuster units. Originally, it was found from experience that up to engine revolutions in the order of 4,000 per minute, hydraulic tappets were highly satisfactory in their action; above that figure there was something lacking in performance. The trouble was in part due to scuffing, but the general adoption of chilled iron tappets in place of steel has overcome this difficulty. Now, engine speeds do not place any restriction on the efficiency of hydraulic valve tappets.

Such is the position of design development to date, and it may be informative to draw comparisons between hydraulic tappets and manually adjusted tappets. There is no doubt that with hydraulic tappets, the valve time is more accurate and is better maintained. With manually adjusted tappets the engine valve may leave or return to its seat at any instant when the cam follower is anywhere along the slope of the cam. In consequence, alterations in valve timing occur as a result of temperature changes. Furthermore, equality in tappet clearance for all valves is practically impossible to attain and maintain by manual adjustment. In contrast, the hydraulic tappet will reproduce the exact timing for which the cams are designed. It will also maintain the timing under all temperature changes on any number of valves to give consistent performance even when conditions vary.

Hydraulic tappets can also have an appreciable effect on engine performance. In fact it has been found that with hydraulic tappets the cam contours can be modified, since no clearance is required, to give an increased engine output up to as much as 10 per cent with uniform performance over lengthy periods. Valve opening may be more rapid and over a longer period than is possible with manually adjusted tappets.

With regard to engine noise, it is well known that manually adjusted tappets are difficult to set for quiet running. Indeed, attempts to obtain quiet operation may reduce engine output or cause rapid valve deterioration. Valve contour and impact between parts separated by clearance are other sources of noisy valve operation. When hydraulic tappets are used, clearance is taken up automatically, and axial contact is ensured between the parts comprising the valve operating assembly. As a result, impact and its resultant noises are greatly reduced, and the engine will have much smoother running characteristics.

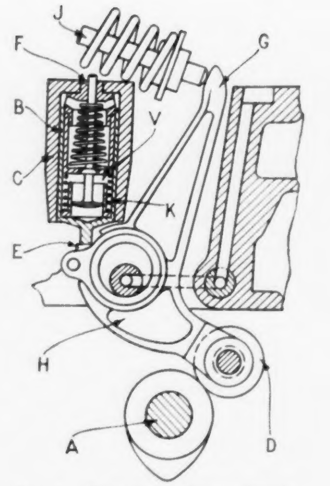


Fig. 7. Packard hydraulic tappet

SEALING POROUS CASTINGS

The Pulsometer Impregnation Process

A RECENT development of interest to the engineering industry is the Pulsometer process of impregnation for the sealing of porous castings. It has now been successfully applied to bronze, gunmetal, steel, cast iron and aluminium castings of all sizes, some of which have included highly finished intricate engine castings costing hundreds of pounds. Suction and delivery covers of pumps weighing over half a ton each have also been treated and stand up, after impregnation, to a pressure of 1,200 lb/in^2.

It is interesting to note that the process was originally conceived as an expedient in the Pulsometer factory for the saving of castings which were shown to be porous under pressure tests. During the war years especially when speed as well as efficiency was essential, many valuable hours were saved by this method whereas if new castings had had to be made the deadline in many cases might not have been met.

In response to outside demands, the service has now been extended to enable castings needing attention to be received from other engineering firms. Through this special process of impregnation thousands of pounds' worth of ferrous and non-ferrous castings have been saved.

Money and labour can be economized if faulty castings can be rescued from the scrap heap and valuable time saved as otherwise new castings would have to be made and the skill and research which had gone into highly finished and expensive pieces of work would be wasted. The Pulsometer Engineering Company can now process castings at their own factory at Nine Elms Iron Works, Reading, or where this is impracticable, as in the case of overseas firms, they are prepared to supply a similar complete impregnation plant to enable manufacturers to carry out the processing direct on the spot.

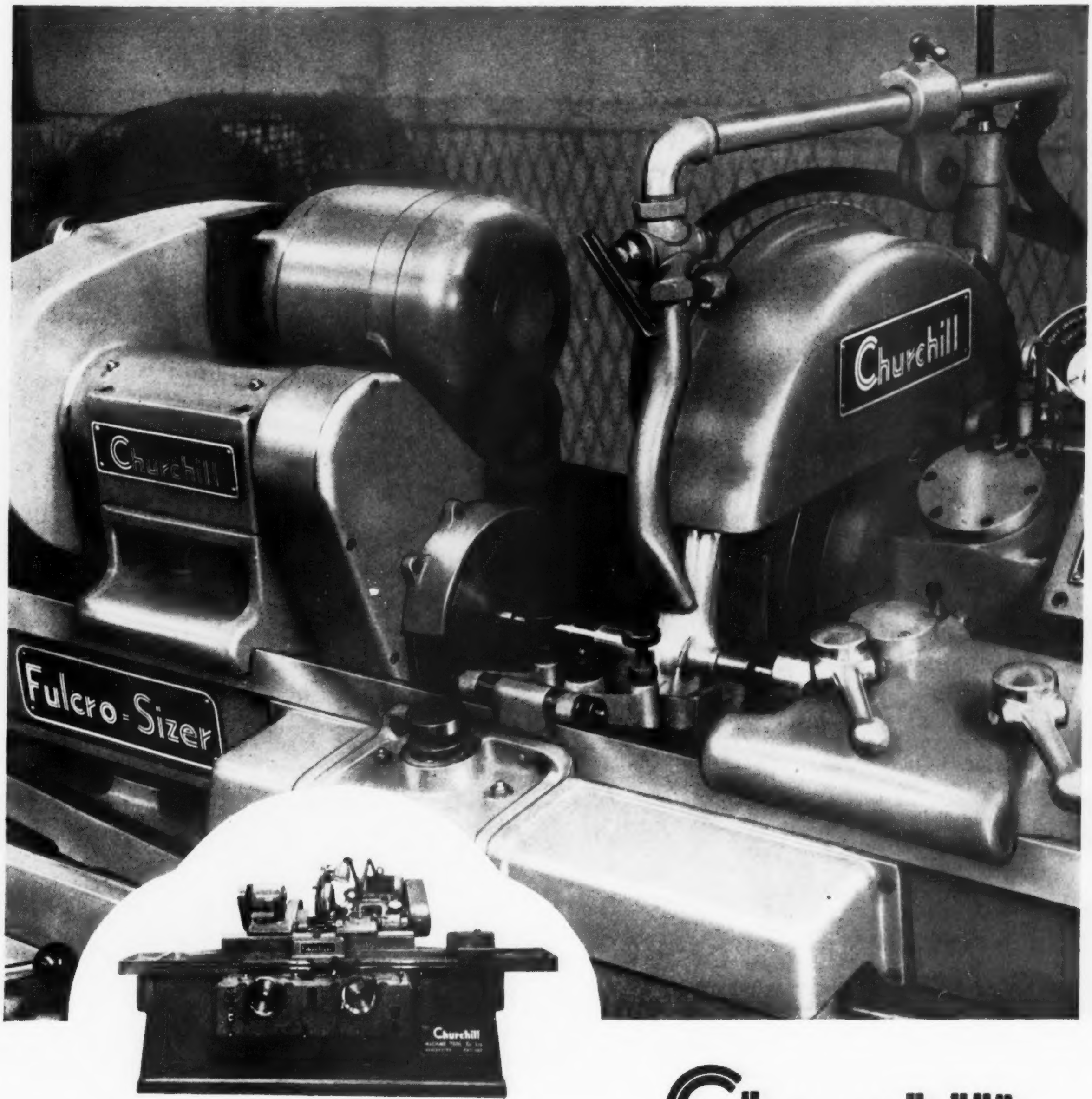
As a matter of interest they have recently received castings from as far afield as Finland as a preliminary to the installation of impregnating plant there. The great value of this particular method of impregnation, which is A.I.D. approved, is due partly to the efficiency of their vacuumizing process and partly to the care with which each subsequent process is conducted. (2039)

Register of Motor Industry

THE fourth post-war edition of the above Register, the Red Book, incorporating all the additions, amendments and deletions made to the 1952 Register, has recently been published, The layout remains unaltered and the introduction again includes certain details of the Distribution Scheme under the provisions of which the former Blue Book was compiled.

The Register lists, under town alphabetical order, some 25,000 companies, firms and persons known to be *bona fide* engaged in the motor industry and all are classified under one or more of forty-nine categories.

Copies are available from the Registers Department of the Society of Motor Manufacturers and Traders, 148, Piccadilly, London, W.1.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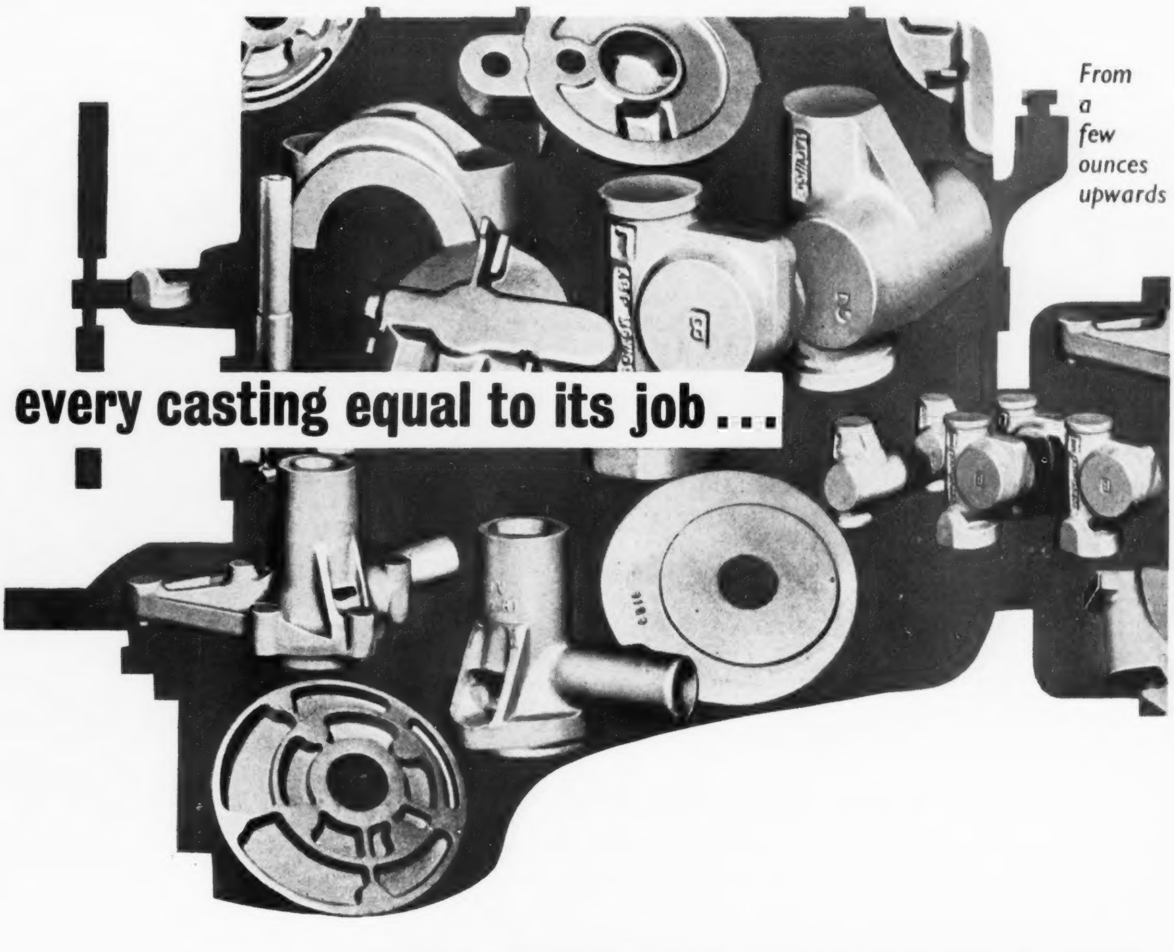
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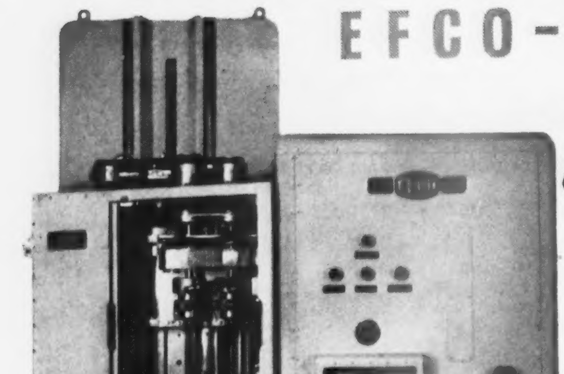

Illustration shows a vertical progressive induction hardening machine for improving torque and wear-resisting qualities of carbon steel axle shafts. Machine treats two shafts at a time and, when powered by a 125kW 10kc M.G set has an output of 60 shafts per hour.

Two such equipments are installed in the Luton Works of Vauxhall Motors Ltd. to whom our acknowledgements are due

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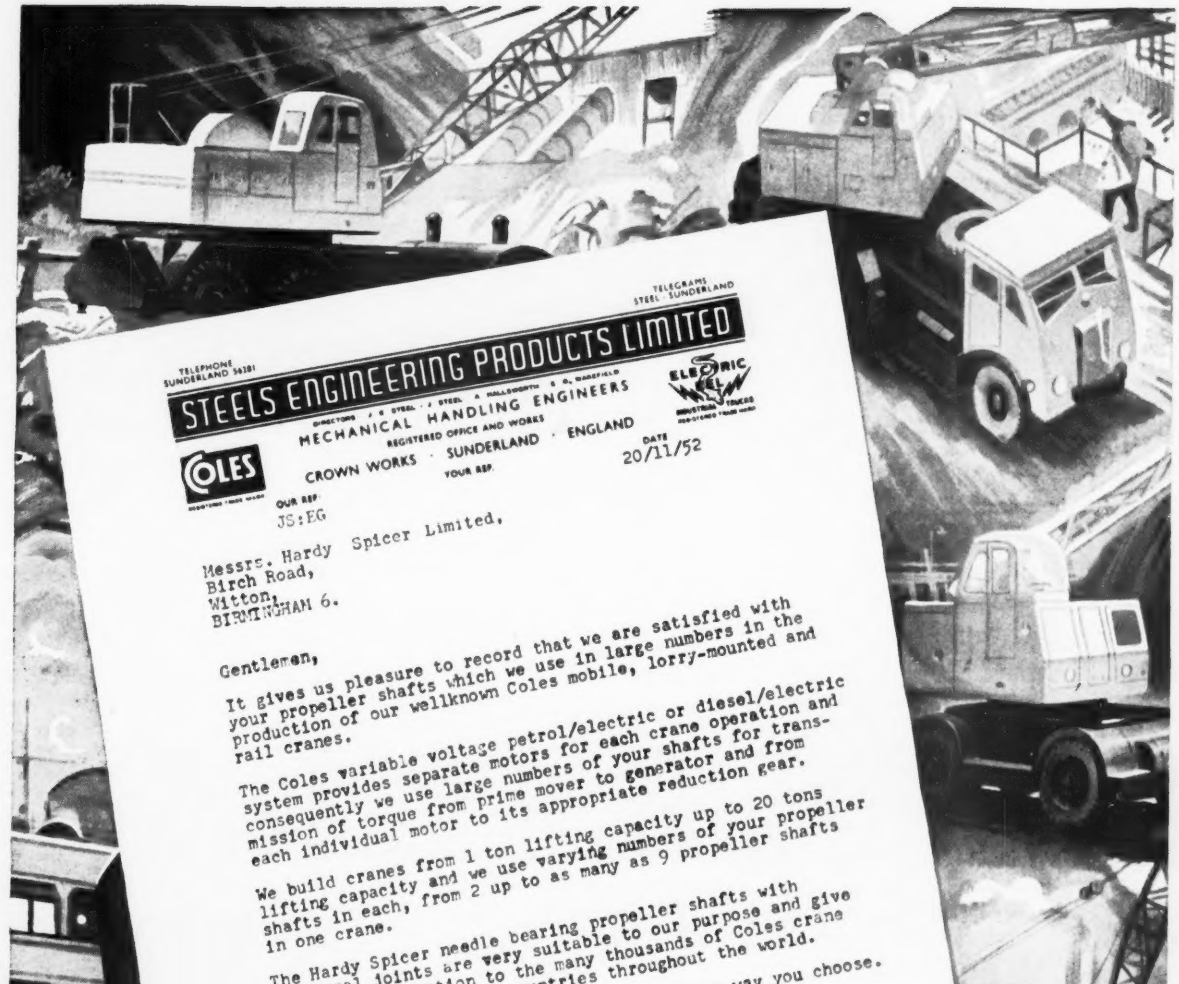

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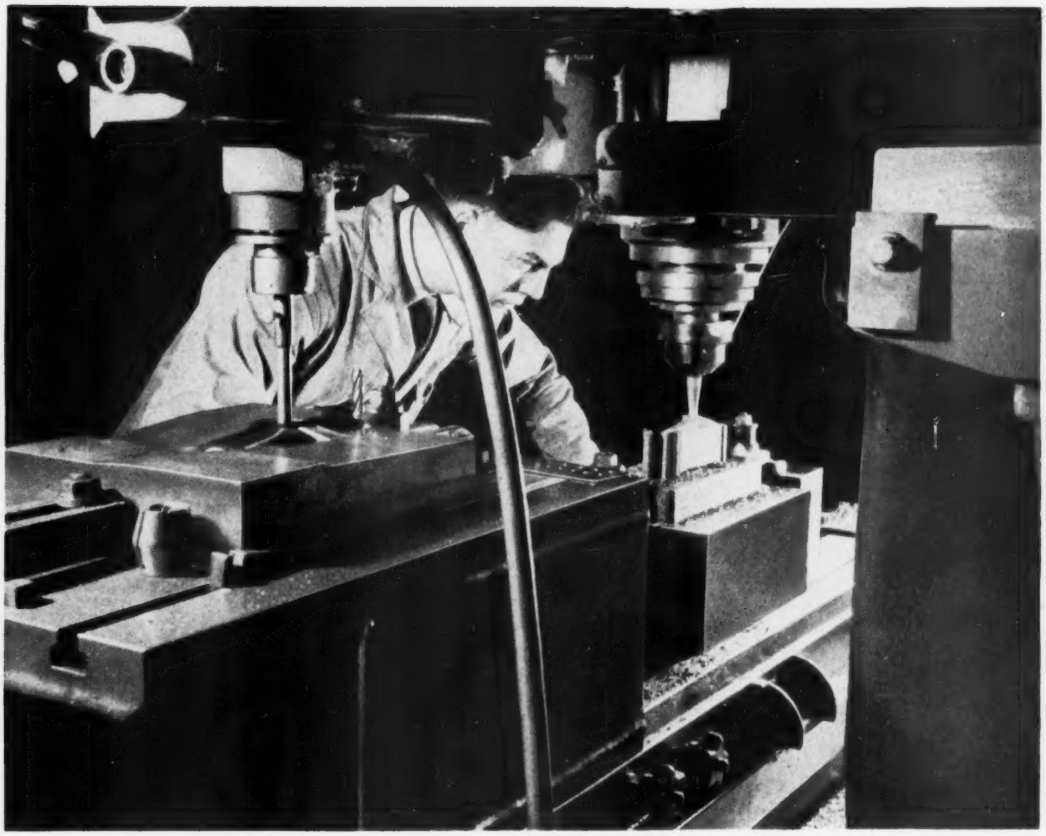

Interesting
. . . but tough

Heat-resisting steels are tough and not easily deformed even at high temperatures. Dies for drop-forging the elaborate shapes of gas turbine blades have a pretty hard job of work to do. Above are dies being machined at Doncasters on a profile copying machine. The lower illustration shows the beginning of marking-out a die block for the drop forging of connecting rods.

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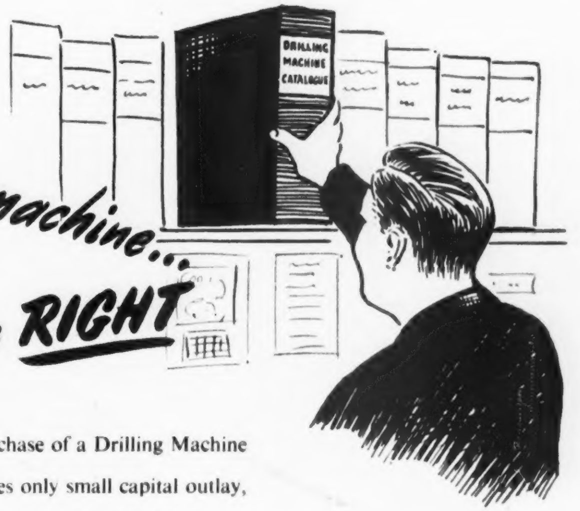

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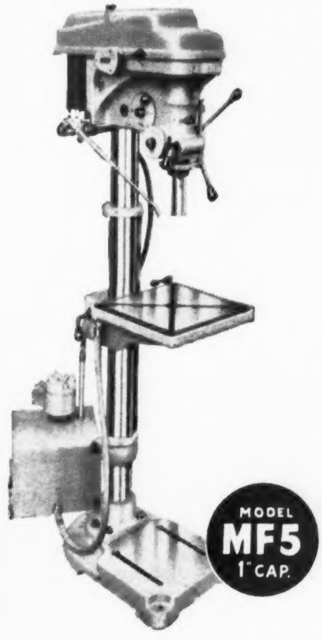

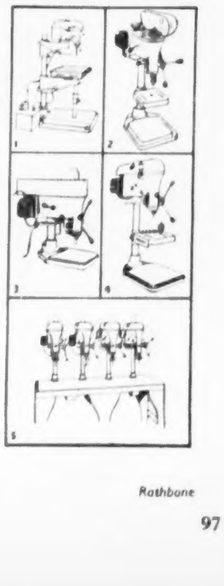

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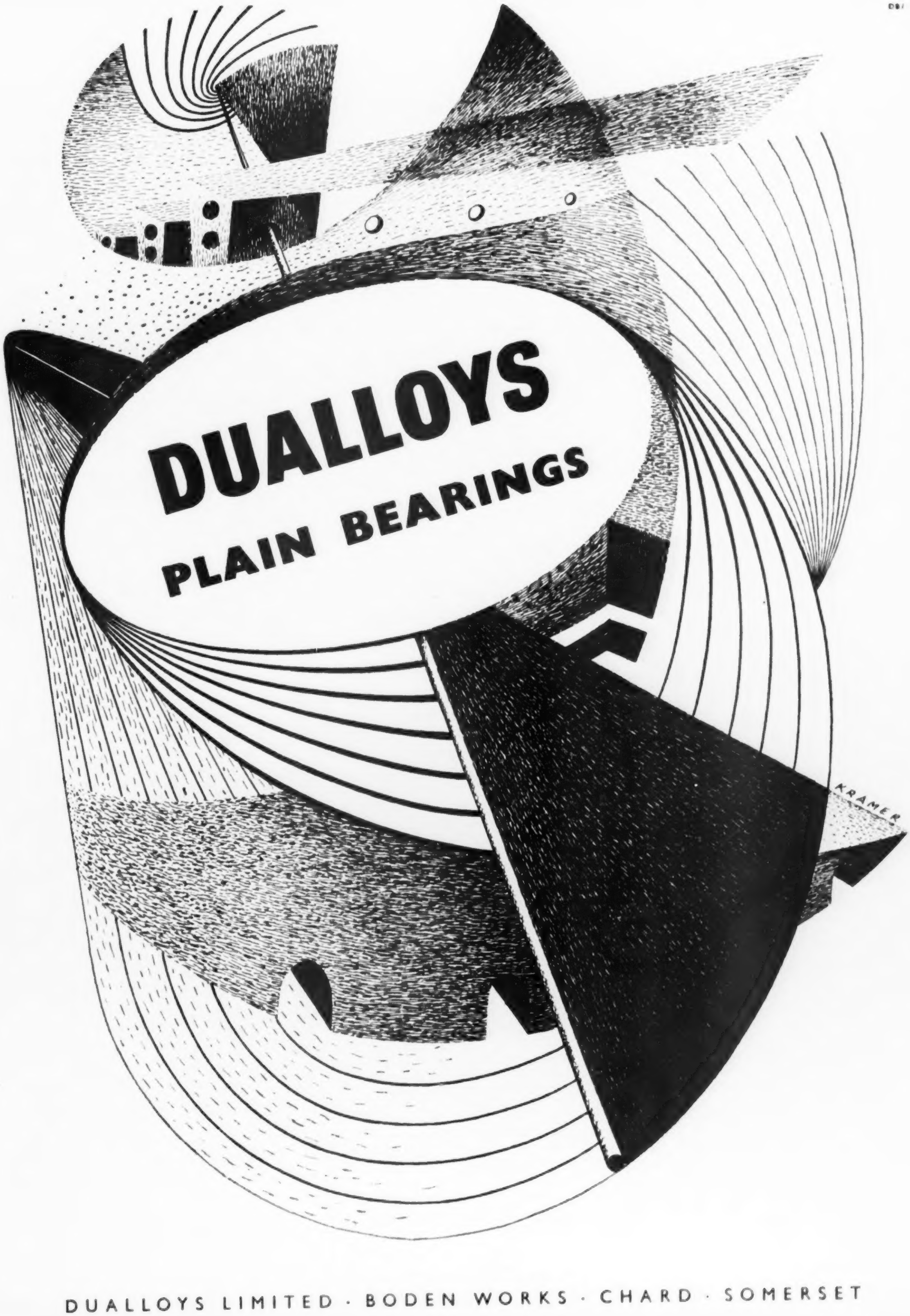
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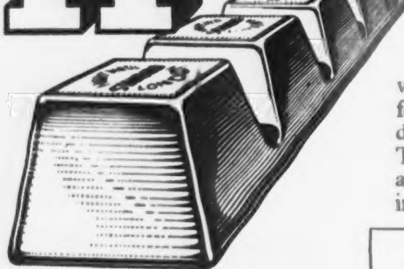

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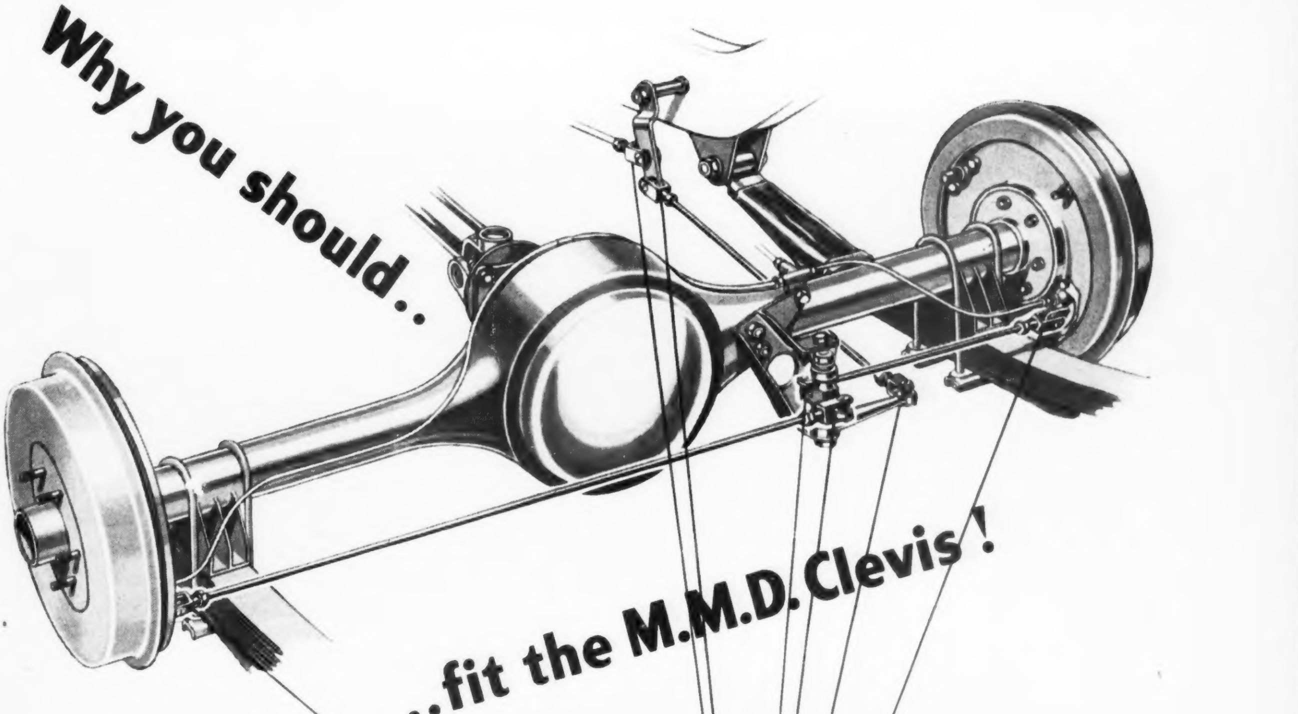

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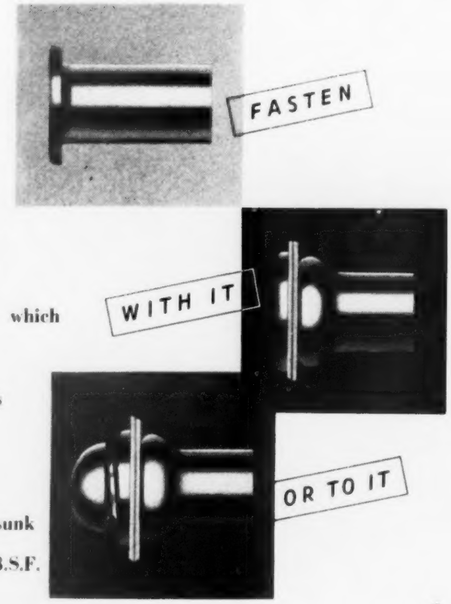

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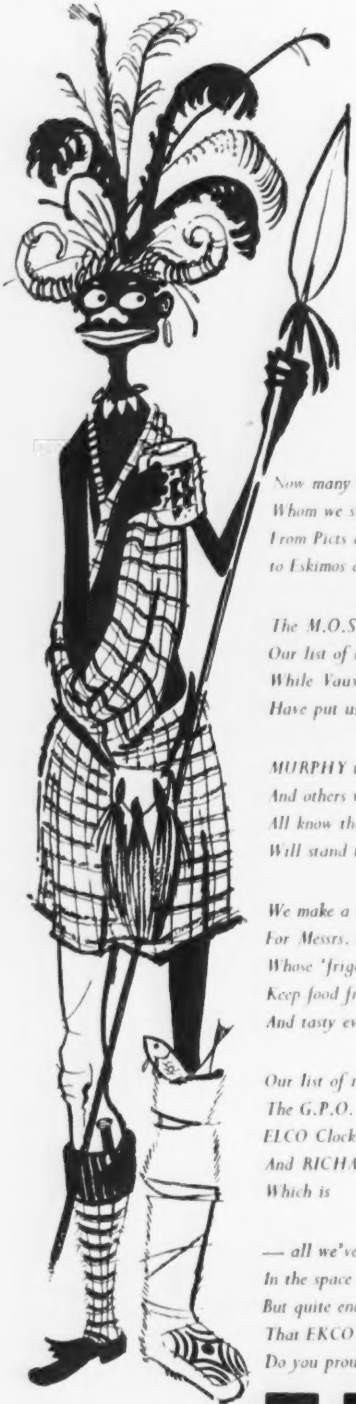
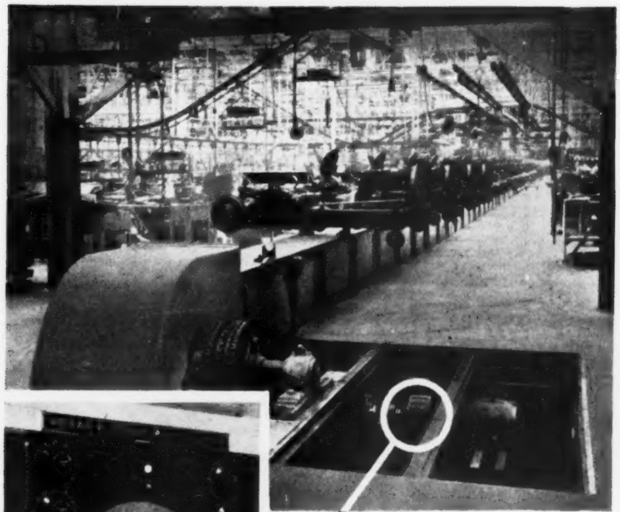
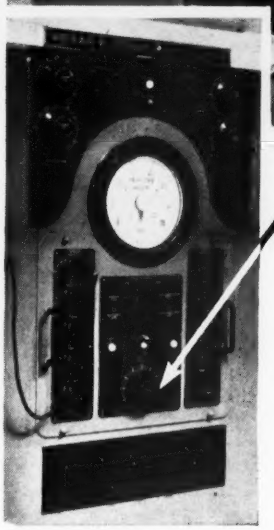




Ferguson Tractor Assembly Track



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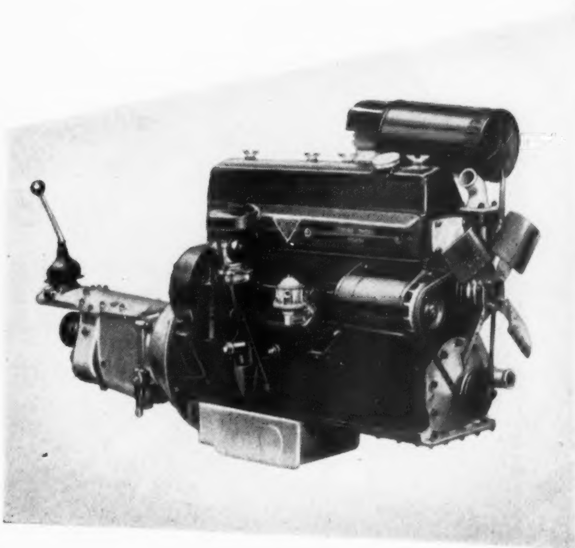

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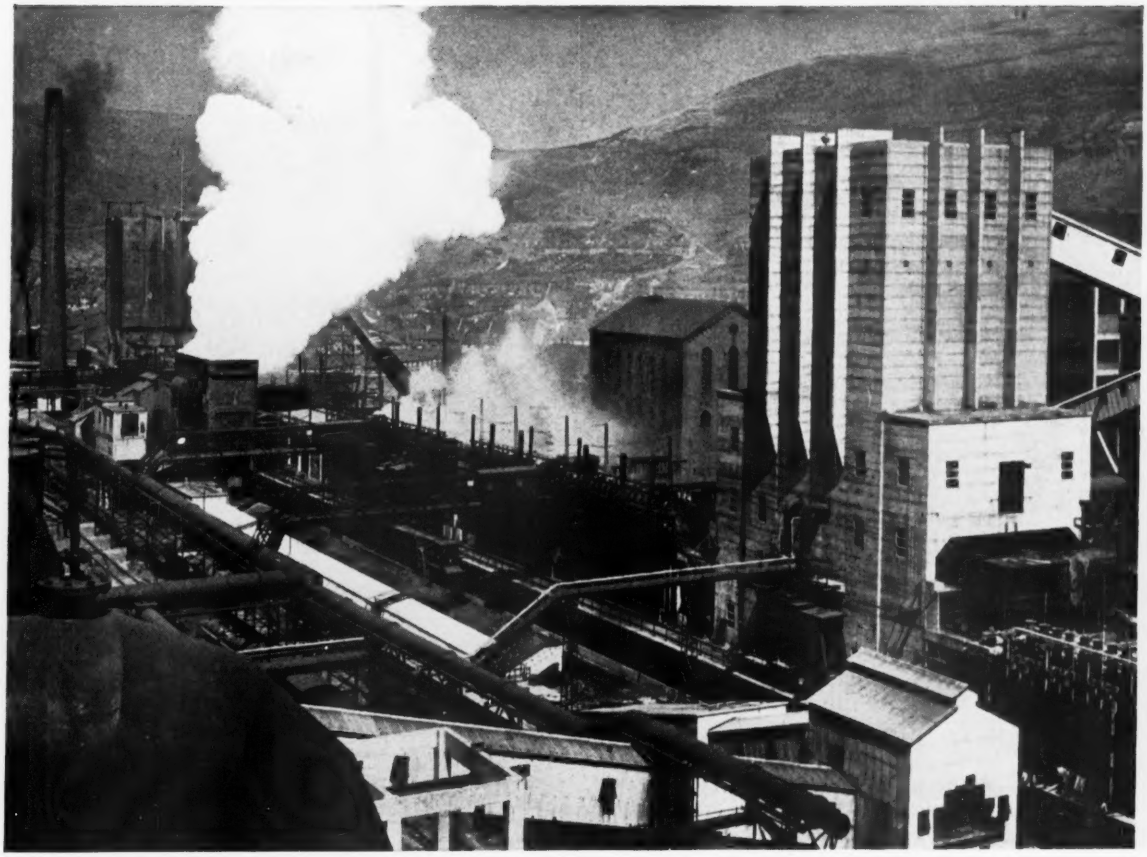

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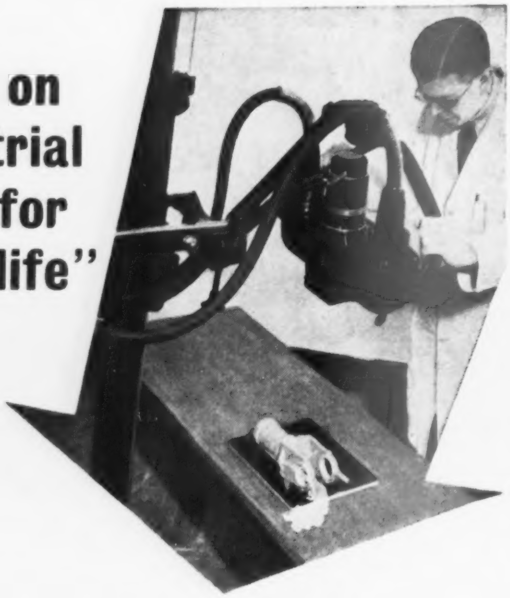

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This is an important occasion; everything hangs on the verdict—not only the life of the component itself but all the other lives that may depend on its soundness.

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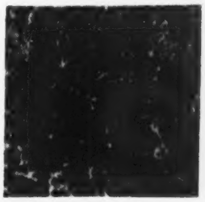
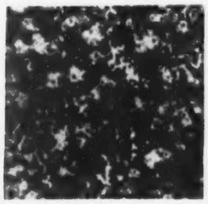
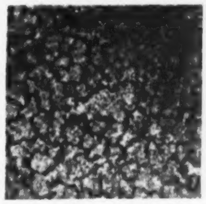
These enlarged photographs of mild steel plates after exposure to the standard Humidity Conditions Test demonstrate the relative degrees of protection afforded by oil media in comparison with RODOL.



Heavy Petroleum Jelly—condition after 7 days, rust attack following original smear tracks.



Paraffin Oil—whole plate deeply rusted after 3 days.



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RODOL W.S.—completely rust-free after 180 days, although plate was rough emeried at outset to accelerate possible rusting.

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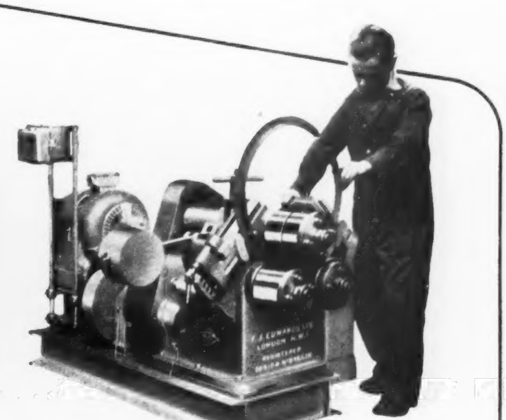

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Pressure die castings in aluminium alloy form the major body components of the "Flamemaster" hand torch illustrated here by courtesy of Messrs. Chance Bros. Ltd. Our customer says: "The aluminium die castings which we have obtained from you for use on this product have proved entirely satisfactory, and are a great improvement on the previous covers. It is in fact true to say that they save us considerable assembly cost due to their accuracy of fit and excellent finish." *Extract of letter from Messrs. Chance Brothers Ltd., Smethwick, Birmingham 40.*

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